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NOISE AND VIBRATIONS CONTROL

A. Stans, G. Silas et al.

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16. Abstract

The many possibilities of modern acoustics find their confirmation in its technological and industrial applications in metallurgy, construction, transportation, electronics, fine mechanics, medicine, recording technology, etc. There scarcely exists any modern technological branch that does not benefit from the aid of acoustics. As a succinct illustration let us note: research on the fatigue of metallic structures by the method of acoustic and ultraacoustic emission, nondestructive testing of the properties of the materials used in different industrial branches, plastic deformation of metals in a macrosonic field, study of the molecular structure of matter with the aid of hypersounds, acceleration of chemical reaction by ultrasonic treatment, acoustic holography, study of magnetoacoustic wave distribution in plasma, ultrasonic defectoscopy, welding of integrated circuits with the aid of ultrasounds, and many others. In most applications the advantages of acoustic methods as compared with "classical" methods are obvious.

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SOME PROBLEMS OF MODERN ACOUSTICS

A. Stan

Acoustics Commission

Keeping pace with the advances of science and technology, /17* modern acoustics is undergoing intensive development from the standpoint of basic research as well as fields of application, with much technological, economical and social spin off. Basic research makes large-scale use of all the modern apparatus that the development of the mathematical and physical sciences places at the disposal of today's researcher. From modern Fourier analysis to information theory, from statistical theories to correlation analysis, the new mathematical research methods are finding a broad field of application in the study of acoustics. Under these conditions the classical physical theories of Kirchhoff, Sommerfeld, Rayleigh or Born are acquiring a more elegant and general formulation so as to comprehend an ever greater number of phenomena. Thus, we have arrived at a deeper understanding of the problems of acoustic wave propagation in the most diverse media, the discovery and theoretical interpretation of new aspects of relaxation, cavitation, dispersion and other phenomena. New branches too have been created, such as nonlinear acoustics and molecular acoustics, and the foundations of quantum acoustics have even been laid.

But the most intensive development refers to the many applications of modern acoustics in practically all spheres of economical, technological and social life [1]. The principal characteristic of this process is the appearance and extension of special branches (Fig. 1) located at the boundary between acoustics and other /18 fields [2].

Analyzing statistical data on the number of papers on acoustics published since 1940, as well as studies published on the occasion of different national and international acoustics congresses, we find that beginning with 1960 a substantial increase has taken place in the proportion of research in the fields of physical acoustics, ultra-acoustics and the fight against noise and vibrations.

This increase can be explained, on the one hand, by the fact that physics and technology can no longer develop without acoustical studies and, on the other, by the specific technological, economical and social implications of acoustics.

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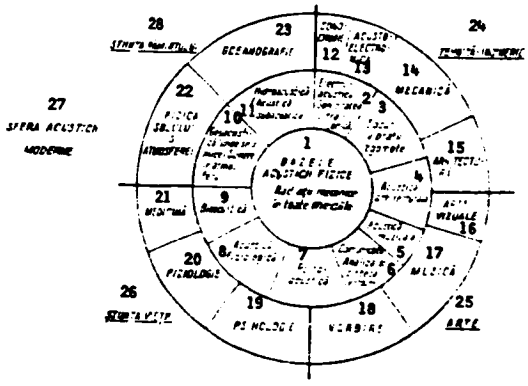

Fig. 1. 1. Bases of physical acoustics: mechanical radiation in all media; 2. Electroacoustics, ultrasonic sonicity; 3. Shocks, vibrations, noises; 4. Architechtural acoustics; 5. Musical acoustics; 6. Communications, speech analysis and synthesis; 7. Psychoacoustics; 8. Pysiological acoustics; 9. Bioacoustics; 10. Geoacoustics (seismic waves), sounds in the atmosphere; 11. Hydroacoustics (underwater acoustics); 12. Sonochemistry; 13. Acoustoelectronics; 14. Mechanics; 15. Architecture; 16. Visual arts; 17. Music; 18. Speech; 19. Psychology; 20. Physiology; 21. Medicine; 22. Soil and atmospheric physics; 23. Oceanography; 24. Technology-engineering; 25. Arts; 26. Life sciences; 27. Domain of modern acoustics; 28. Earth sciences.

Underlying the development of these new branches is basic physical mathematical research correlated with particular disciplines (engineering, biology, medicine, geophysics, etc.) as well as ever more perfected means of experimental investigation.

In order to amplify experimental research it was necessary, in the first place, to extend the frequency range considerably beyond the audible, to the infrasonic and ultrasonic as well as the hypersonic range.

In addition to natural /1
infrasounds (earthquakes, winds, volcanic eruptions, tides) there also exist many sources of industrial infrasounds (diesel engines, some types of ventilators with a reduced number of rpm, ceramic kilns, flight of hypersonic aircraft, etc.) in whose spectrum there are to be found infrasounds of rather high intensities that can produce considerable damage.

The noxious character of infrasounds requires special measures to be taken directly at the source that generates them.

Present-day norms fix the limits of the noxious effects of noise only for the audible range without taking into consideration the harmful action of infrasounds, ultrasounds and hypersounds. So here is a new field of research.

As regards the field of hypersounds, experiments performed with frequencies in excess of 100 gigahertz [4] have brought to light their action in the physics of condensed matter:

-- in the field of acoustic paramagnetic and electron-phonon resonance;

-- in the excitation of condensed matter (spin waves, electromagnetic waves, study of electron plasma, etc.), with critical fluctuations if the medium undergoes a change in phase;

-- in the field of fluids -- in the broad acceptation of studies that aim at the study of relaxation or propagation phenomena (fluid crystals) as well as phase-change properties.

In addition to an extension of the frequency range, scientific and technological advances have brought in their train a considerable increase in levels of acoustic intensity. Thus, at the present time we have attained acoustic intensities up to 10^5 w/mm^2 as compared with the 10^3 w/mm^2 that represents the maximum acoustic intensity supported by the human ear, corresponding to 120-130 dB.

Such extremely high intensity levels [5], besides raising new problems concerning the generation, measurement and study of the action of such intensities on the organ of hearing as well as on other organs of living beings, raise problems concerning the very hypotheses and limits of linear acoustics, so that it is becoming necessary to introduce new hypotheses leading to the creation of nonlinear acoustics [6]. /2(

Undergoing development at the present time is a modern apparatus for the generation, picking up and recording of acoustic waves and for the study of their effects on the most diverse media. In other words, the broadening of the experimental basis of investigation has in view the study of problems connected with source and sound wave properly speaking as well as problems connected with the interaction of the latter with material media.

Already built on an industrial scale are many devices and machines that can be used in research and production tending toward the automization of machines.

The many possibilities of modern acoustics find their confirmation in its technological and industrial applications in metallurgy, construction, transportation, electronics, fine mechanics, medicine, recording technology, etc. There scarcely exists any modern technological branch that does not benefit from the aid of acoustics. As a succinct illustration of some of the problems studied by acoustics in some of the above-mentioned fields, let us note: research on the fatigue of metallic structures by the method of acoustic and ultraacoustic emission, nondestructive testing of the properties of the materials used in different industrial branches, plastic deformation of metals in a macrosonic field, study of the molecular structure of matter with the aid of hypersounds, acceleration of chemical reactions by ultrasonic

treatment, acoustic holography, study of magnetoacoustic wave distribution in plasma, ultrasonic defectoscopy, welding of integrated circuits with the aid of ultrasounds, and many others. In most applications the advantages of acoustic methods as compared with "classical" methods are obvious from the standpoint of efficiency as well as profitability.

In a report presented to the VI International Congress on Acoustics in Tokyo [7] it is shown that among factors that have contributed to the modernization of industry and the creation of new machines and industrial equipment, acoustics has played an important role. /2

Its applications and technological implications attest to the advanced stage of development of modern acoustics and its interconnection with other scientific and technological branches.

In the context of these connections a place apart is occupied by those fields which have particularly important social implications for the daily life of the modern human collectivity. The rapid increase in noise sources, especially in conurbations raises -- as everybody knows -- major noise pollution problems.

From the noise of means of conveyance (cars, buses, aircraft) to the noise of great industrial enterprises (forges, weaving mills, spinning mills, etc.), the multitude of noise pollution sources leads to a reduction in working capacity and deteriorates the health of people, sometimes in an irreversible manner. As a consequence,qualified personnel, whose professional training requires a long time, are prematurely dismissed from production. Moreover, vibrations whose physical characteristics exceed certain limits have harmful effects not only on man, but also on machine parts and construction elements, and can lead to their degradation and even destruction.

Thus, the material losses directly or indirectly due to noise and vibrations come to considerable amounts every year. The problems that are raised refer to the quantitative determination of the damage done and its implications, as well as to the measures that must be taken. Today, the fight against noise and vibrations constitutes an important branch of environmental science, which the UNO regards as one of the central problems of mankind in the near future.

That is why we are stressing in this connection the fact that the fight against noise pollution involves certain additional investments that often meet with the unjustified opposition of responsible parties.

There can be no doubt that this is a harmful attitude that prevents sound long-term analysis. Besides the other conditions of comfort that society creates for man, acoustic comfort must /2

undoubtedly become one of the principal components to assure optimum recovery of strength for physical and intellectual labor.

Owing to this fact, the efficiency with which funds earmarked for the creation of acoustic comfort are invested must be worked out in a more realistic manner.

By extending basic research and its material bases, modern acoustics is assuming a double character: on the one hand, it is developing as an independent science with its own means and methods of investigation and, on the other, it is emerging as a typical interdisciplinary science in which two or more distinct fields overlap.

But this characteristic -- which, by the way, is also encountered in other modern sciences -- raises the problem of collaboration between researchers from different specialties. This collaboration requires not only harmonious completion of knowledge in the respective limiting fields, but also judicious coordination of research. There thus emerges the necessity of grappling with a great number of problems with a view to discovering the possibilities afforded by the interdisciplinary field of acoustics. Considering the extreme specialization that characterizes the modern researcher, a special role in forming a sound general view of modern acoustics belongs to the learning process. Owing to the dual nature of acoustics as a specific and interdisciplinary science, not even its rudiments can be grasped by improvisations, but only by sound preparation at the university and post-university level. Much more extensive university courses than exist at present would have to assure a broad initiation into the many theoretical aspects of acoustics, while post-university courses would create the conditions for stricter specialization. The variety of problems raised by modern acoustics dictates the development of /2
acoustics departments in universities and technical institutes as well as the establishment of specialized institutes. In this manner, a closer connection could be brought about between learning, research and production, which would be useful not only to the development of the science, but also to the entire national economy.

In achieving these objectives, the experience of other countries may prove to be particularly useful. Thus, in many countries there exist learning and research establishments designed exclusively for the problems of acoustics, such as: the institutes of acoustics of Moscow, Leningrad and Peking, the Institute of Acoustics and Vibrations of Southampton University in England, the Institute of Acoustics under construction in Dresden, etc, or centers of acoustic research (RPPRSC, France; RPU, Austria; Italy, etc.).

At present, over 18 acoustical societies with approximately 3,000 members are active in Europe alone, and in America there are over 4,200 members of acoustical societies. While during the

First World War there were only 300-400 persons who dealt with the problems of acoustics, at the present time there are over 25,000 researchers engaged in this important branch of modern science.

The special importance that is accorded to the study and development of modern acoustics is also reflected in the many national congresses on acoustics that are periodically organized in practically all countries.

Of particular interest are the research results appearing in papers published by different international congresses and, especially, the congresses organized every three years by the International Acoustics Commission under the auspices of UNESCO's International Union of Pure and Applied Physics (IUPAP). A continuous increase in the number of participants and papers can be observed. While at the 1959 Stuttgart International Congress 800 participants presented 350 papers, 1,000 participants presented 365 papers at Copenhagen in 1962, and 1,400 participants presented 700 papers at Budapest in 1971 (Fig. 2). /2

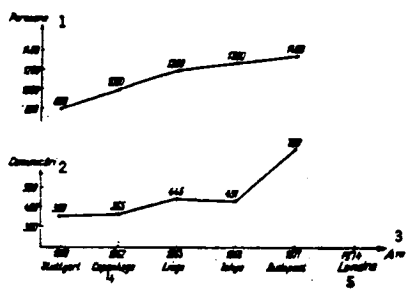

Fig. 2. 1. Persons; 2. Papers; 3. Years; 4. Copenhagen; 5. London.

In our country a particular contribution to the development of acoustics has been made by the Acoustics Commission of the Academy of the RSR, which for over 18 years has been the cynosure of all Rumanian activity in the different branches of acoustics. Thus, among other things, it organized courses, symposia, conferences and congresses (including the 1969 Congress, in which many specialists participated), promoted the introduction into practice of research results, printed publications on acoustics, etc.

Owing to the complexity of the problems of principle and organization that modern acoustics raises, an analysis, be it ever so succinct, of all its directions of development is difficult to undertake within the limits of a short report. Such an analysis would also be rendered difficult by the fact that acoustics is, at the present time, a particularly mobile science undergoing continuous development, sometimes not lacking in the unforeseen. Ever newer applications are cropping up, as well as new and different research possibilities.

For this reason we have confined ourselves in the present report only to some aspects that we deemed to be the most representative.

Summary

/26

The present paper discusses some aspects of modern acoustics, the science of mechanical radiation. The following are highlighted:

-- the multidisciplinary and interdisciplinary character of acoustics;

-- its scientific, technological, economical and social implications, as well as the role of acoustics in creating new machines and equipment and improving the quality of products;

-- the tendency to extend research beyond audible frequencies, as well as to extremely high acoustic intensities, which requires the development of a nonlinear acoustics;

-- the necessity of creating new materials and discovering new methods of reducing noise and vibration levels;

-- the development of automated measures;

-- the necessity of accentuating the training of specialists in acoustics at all levels;

-- the necessity of close international collaboration between specialists in terms of acoustics societies and commissions.

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STUDY OF VIBRATIONS PRODUCED BY A VIBRATING BEAM USED FOR VIBRATING CONCRETES

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1. Introduction

/2

For surface compacting, a number of vibratory implements are used, such as vibrating plates and beams. These implements are of simple design and are employed especially in road building. The action of vibrations on concrete leads to a compacting of the material together with an improvement in its mechanical properties and a commensurate smoothing of its surface.

Most concrete compacting implements are manually operated, so that the vibrations transmitted to the handle must not be allowed to reach harmful levels.

In the present study we discuss some theoretical and experimental results obtained with a vibrating beam built in Rumania. But nothing is to prevent our study methods from being applied to other manually-operated vibratory implements.

2. How the Beam Works

In order to explain the beam's vibratory motions we must briefly describe its operating principle. The component parts of the vibrating beam used in concrete compacting are shown in Fig. 1.

Fastened to the beam, properly speaking, (1) made of hard wood, is the inertial generator (2) with eccentrics that impart to the beam vibratory motions in two directions.

The beam's horizontal motion smooths the concrete mass (3) to the height of the stop beams (4) propped up by the ground while vertical motion compacts the concrete.

The beam is manipulated by means of the handle (5), which is fastened to the beam by means of the rubber pads (6) that play the role of reducing vibrations at the end of the handle. /2

The vibration generator (2) is electrically activated by an asynchronous three-phase motor with a fixed number of revolutions (3,000 rpm). As a consequence, only the vibration amplitude characterizes vibration conditions for compacting.

Generally, the conditions of motion are chosen in such a manner that the beam will be periodically unfastened from the stop. The shocks that occur as the beam drops against the stop must not be transmitted to the handle.

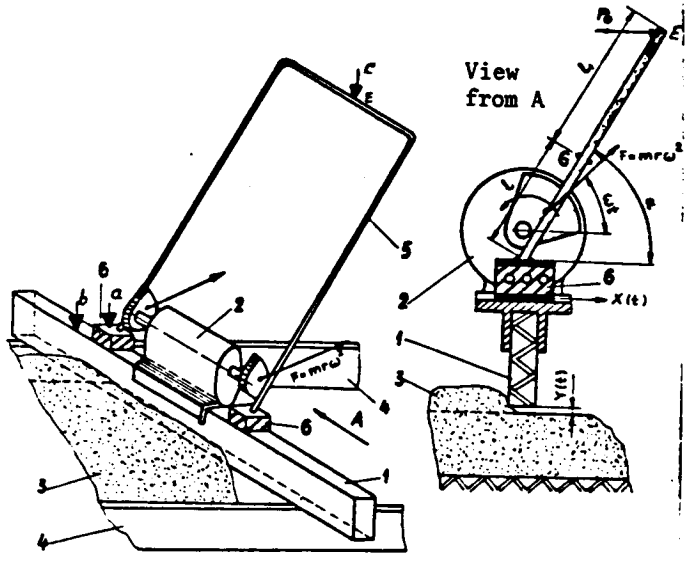

Fig. 1

As has been shown, the vibratory motions are transmitted to the handle by means of rubber pads that play the role of shock- and vibration-absorbers. For this reason, the elastic characteristics of the pads must be chosen in a judicious manner.

3. The Beam's Shock Vibration Motions

/2

As the beam moves it must be periodically unfastened from the stop. The beam's shock vibration motions can be studied in terms of a simplified dynamic model (Fig. 2) in which the mass M is the mass of the beam-vibrator assembly. As compared with this mass, the system's other masses can be neglected. The mass M is acted upon by the vibration generator with the eccentric masses m_0 and the eccentricity r, as well as by a horizontal force P_0 assumed to be constant, applied by the operator. In this way, the mass M will be acted upon by forces
$P_x = P_0 + m_0 r\omega^2 \sin \omega t$ and
$P_y = m_0 r\omega^2 \cos \omega t$.

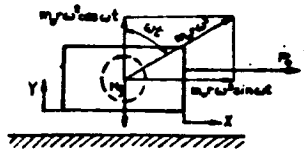

Fig. 2.

The differential equations of motion of the mass M will be:

$$M\ddot{x} = P_0 + m_0 r\omega^2 \sin \omega t$$
$$M\ddot{y} = m_0 r \omega^2 \cos \omega t - Mg \qquad (1)$$

Likewise assumed known are the resetting ratios R and λ of the velocities tangential and normal to the impact ($0 \leq R \leq 1$, $0 \leq \lambda \leq 1$).

Designating by t_k ($k = 1,2,\ldots,$) the moment of a certain impact, for the velocities at the beginning and end of the impact [3] we can write the following equations:

$$\dot{x}'_k = (1-r)\dot{x}_k \tag{2}$$

and

$$\dot{y}'_k = -R\,\dot{y}_k \tag{3}$$

The motion of the mass M can be decomposed into two components corresponding to the X and Y coordinates taken separately. The vertical component determines the character of the motion [1]. The law of motion in the range (t_k, t_{k+1}) between two successive impacts will be

$$Y(t) = \frac{m_0 r}{M}\left[\cos\omega t_k - \cos\omega t - \omega(t-t_k)\sin\omega t_k\right] + \dot{Y}(t_k)(\\ -\frac{g}{2}(t-t_k)^2. \tag{4}$$

Inasmuch as what interests us is periodic motions with a period $2\pi/\omega$, we must consider $\dot{Y}(t_k) = -v$. Thus, from the conditions between two successive impacts we can, by using Eq. (4), get: /3(

$$\cos\omega t_k = \frac{gr}{\omega}\,\frac{M}{m_0 r}\,\frac{1-R}{1+R}\,,\quad v = \frac{2gr}{1+R} \tag{5}$$

If the values of t_k and v are introduced into the solution of Eq. (4), we can set approximately $Y = Y_0 \sin\omega t$.

We can proceed in the same manner in the case of horizontal motion. To study the motion we will integrate Eq. (1), taking into account Eq. (3). In this manner we will arrive at a solution of the form $X = X_s + X_0 \cos\omega t$.

4. Transmission of the Beam's Vibratory Motions to the Handle

The handle of the vibrating beam has a mass by far smaller than that of the beam-vibrator assembly, so that the vibrating system consisting of handle and rubber pads does not influence the beam's vibrations.

The rubber pads are considered to be elastic restraints with the rigidities k for rotation, k_1 along the x direction and k_2 along the y direction. The differential equations of the handle's plane motions as compared with the position of equillibrium are

$$\begin{aligned} m\ddot{x} + k_1 x - \ddot{\varphi} m l \sin\alpha &= k_1 X(t) \\ m\ddot{y} + k_2 y + \ddot{\varphi} m l \cos\alpha &= k_2 Y(t) \\ J\ddot{\varphi} + k\varphi + k_1 x l \sin\alpha - k_2 y l \cos\alpha &= k_1 X(t) l \sin\alpha - k_2 Y(t) l \cos\alpha. \end{aligned} \tag{6}$$

in which x,y and ϕ determine the system's position.

Integrating Eq. (6), we get the laws of motion

$$\varphi(t) = ml \frac{1}{J + ml^2\left[\frac{\sin^2\alpha}{1-\left(\frac{\omega}{\omega_1}\right)^2} + \frac{\cos^2\alpha}{1-\left(\frac{\omega}{\omega_2}\right)^2}\right]} \left[\frac{\sin\alpha}{1-\left(\frac{\omega}{\omega_1}\right)^2} X_0 \cos\omega t - \right. \tag{7}$$

$$\left. x(t) = \frac{1}{1-\left(\frac{\omega}{\omega_1}\right)^2} X_0 \cos\omega t - \varphi \frac{\left(\frac{\omega}{\omega_1}\right)^2}{1-\left(\frac{\omega}{\omega_1}\right)^2} l \sin\alpha \qquad y(t) = \frac{1}{1-\left(\frac{\omega}{\omega_2}\right)^2} \right]$$

where we have designated $\omega_1^2 = \frac{k_1}{m}$; $\omega_2^2 = \frac{k_2}{m}$; $\omega_\varphi^2 = \frac{k}{J}$.

In these equations we have neglected $\omega_0 = 2\pi\nu_0 = 2\pi \cdot 2 \cdot 12{,}4 = 77{,}8\ s^{-1}$ /3
as compared with $\omega = 2\pi f = 2\pi \cdot 50 = 314\ s^{-1}$.

Thus, we can move on to finding the vibrations at any point E of the handle by means of the following equations:

$$\begin{aligned} x_E(t) &= x(t) - \varphi(t)\,(l + l_1)\sin\alpha \\ y_E(t) &= y(t) + \varphi(t)\,(l + l_1)\cos\alpha \end{aligned} \tag{8}$$

In Eq. (8) the angle $\phi(t)$ was considered to be small in comparison with the angle α given by the handle's static equilibrium.

It can be seen from Eq. (7) that the minimum values of x(t) and y(t) are smaller for the ratios ω/ω_1 and ω/ω_2 than for soft pads. These pads must be rigid enough not to hamper manipulation of the beam, which can no longer be controlled in the case of very soft pads. On the basis of Eqs. (7) and (8) we can influence the vibrations at the gripping point E by altering the position of the center of gravity G along the bar.

5. Experimental Verification

For experimental verification of our theoretical study we mounted three transducers (Fig. 1): a -- vertically to the handle at the restraining point; b -- vertically to the vibrating beam; c -- vertically to the handle in the gripping zone.

In the chosen case we found that the rubber pads do not live up to expectations because the amplitude at point a of the handle y_0 = 1.35 mm is greater than on the beam, Y_0 = 0.35 mm, so that in the gripping zone y_E = 0.3 mm, which shows that we can have

vibration amplification effects at the handle. Fig. 3 shows vibrograms recorded for these motions.

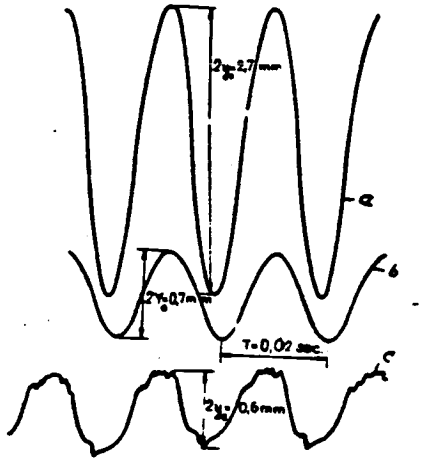

Fig. 3.

Summary

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For compacting concretes in building, vibrating beams are used. The vibrations are generated by inertial vibrators, and the beam is normally displaced by the operator by means of a handle that is elastically fastened to the beam by means of rubber pads. These pads are meant to attenuate transmitted vibrations.

The paper deals with vibrations transmitted to the operator, taking into account the beam's shock vibration motions. In the first part the steady state motion of a dynamic beam pattern is studied, and in the second part the results of experimental tests with existing equipment are presented.

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A MATHEMATICAL MODEL FOR STORIED BUILDINGS SUBJECTED TO AUTOMATIC LOOM STRESSES

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Introduction

In the textile industry there exist storied weaving mills that vibrate powerfully as a result of the technological process. In addition to the harmful effect that vibrations can have on workers, they often also have a considerable effect on buildings. /3

The present paper analyzes the action of the forces produced by the technological process considered as acting at every level of the building on the assumption of harmonic forces as well as in the case of their random variation.

1. Differential Equation of the Structure's Vibrations

The storied building was regarded as a beam fixed at one end with a uniformly distributed load p and, at each level, concentrated loads m_i (i=1...k) (Fig. 1).

Fig. 1.

On this assumption, the differential equation of the forced transversal vibrations [1, 2] is

$$EI\frac{\partial^4 y}{\partial x^4}+\left[p+\sum_{i=1}^{k} m_i\,\delta(x-x_i)\right]\frac{\partial^2 y}{\partial t^2}=g(x,t) \tag{1}$$

with limiting conditions

$$y(0)=0;\quad \frac{\partial y(0)}{\partial x}=0;\quad \frac{\partial^2 y(l)}{\partial x^2}=0;\quad \frac{\partial^3 y(l)}{\partial x^3}=0 \tag{2}$$

where: I is the section's geometrical moment of inertia; δ is the Dirac distribution; g(x,t) is the equivalent load corresponding to the forces $F_k(t)$ that act at the levels 1,2,...k, expressed by

$$g(x,t)=\sum_{i=1}^{k} F_i\,\delta(x-x_i)\left[H(x)-H(x-l)\right] \tag{3}$$ /

and H is the Heaviside function.

The wave equation of the proper vibration modes is

$$EI\,y^{IV}(x)-\omega^2\left[p+\sum_{i=1}^{k} m_i\,\delta(x-x_i)\right]=0 \tag{4}$$

Using the Laplace transform, taking into account the limiting conditions of Eq. (2), with $\lambda=\sqrt[4]{\frac{\omega^2 p}{EI}}$, ω representing the proper values,

we get

$$y(x) = \frac{\Delta}{\beta_0} \sum_{i=1}^{k} m_i \cdot y(x_i) Y_i(x) \tag{5}$$

where

$$Y_i(x) = \varphi_1(\lambda, \ell, x_i) V(\lambda, x) + \varphi_2(\lambda, \ell, x_i) V(\lambda, x) + T(\lambda, x - x_i) H(x - x_i) \tag{6}$$

and S, T, U, V are Crilov functions.

If in Eq. (4) we replace, successively, $x = x_i$ (i=1,2,...k), we get a homogeneous system of algebraic equations with the determinant Δ. The equation of the proper pulsations is $\Delta = 0$. For a solution of the differential equation of the form

$$y(x,t) = \sum_{n=1}^{\infty} y_n(x) f_n(t) \tag{7}$$

the differential equation of the forced vibrations corresponding to the second vibration mode is

$$f_n(t) + \omega_n^2 f_n(t) = \frac{1}{N_n} \int_0^{\ell} y_n(x) q(x,t)\, dx \tag{8}$$

where $N_n = \int_0^{\ell} [\mu + \sum_{i=1}^{k} m_i \delta(x - x_i)] y_n^2(x)\, dx$ is the generalized mass. On the assumption of weak viscous damping, or when the particular conditions derived by Rayleigh and generalized by Caughey [6] are satisfied, also inclued in Eq. (8) is a term that characterizes the damping force.

2. The Case of Harmonic Forces

In this case, the stationary solution of Eq. (8), as well as of the corresponding equation on the assumption of viscous damping, is given by the Duhamel integral

$$f_n(t) = \int_0^{\infty} W_n(z) F(t - z)\, dz \tag{9}$$

where W_n is the unitary response function due to a load of the unitary pulse type.

For the case of a harmonic force $F(t) = F_0 e^{i\omega t}$, the stationary solution is of the form $f_n = \alpha_n F_0 e^{i\omega t}$ where $\alpha_n(\omega)$ and $\alpha_n(i\omega)$ are, respectively, the real and the complex response functions of Eq. (8). /3

For the n-th vibration mode

$$\alpha_n = \int_0^{\infty} W_n(z) e^{-i\omega z} \tag{10}$$

we get

$$\alpha_n(\omega) = \frac{\gamma_n(x_i)}{N_n(\omega_n^2-\omega^2)} \; ; \quad \alpha_n(i\omega) = \frac{\gamma_n(x_i)}{N_n(\omega_n^2-\omega^2+2i\xi_n\omega_n\omega)} \tag{11}$$

where ξ_n is the relative coefficient of viscous damping. In the case of a harmonic force, Eq. (7) has the form:

$$y(x,t) = \alpha_{xx_i} F_0 e^{i\omega}$$

where α_{xx_i} is the response function corresponding to any point x when the ^{i} harmonic force acts at point x_i:

$$\alpha_{xx_i}(\omega) = \sum_{n=1}^{\infty} \frac{\gamma_n(x)\gamma_n(x_i)}{N_n(\omega_n^2-\omega^2)} \qquad \alpha_{xx_i}(i\omega) = \sum_{n=1}^{\infty} \frac{\gamma_n(x)\gamma_n(x_i)}{N_n(\omega_n^2-\omega^2+2i\xi_n\omega_n\omega)} \tag{12}$$

For equivalent forces of the form of Eq. (3) with

$$F_i(t) = F_{0i} e^{i\omega t} \qquad i = 1, 2, \ldots k \tag{13}$$

the solution of Eq. (8) has the form

$$y(x,t) = e^{i\omega t} \sum_{i=1}^{k} F_{0i}\alpha_{xx_i} \tag{14}$$

and if the excitatory pulsation ω is close to one of the proper pulsations, then the series can be satisfactorily approximated by a single term of the series.

3. The Case of Random Forces

It can be assumed that the random forces are represented by the stationary functions and that the corresponding displacements are acting in the direction of the forces. If the stationary random force F_i of spectral density $S(F_i)$ is acting at the level x_i, bearing in mind that the differential equations of the proper modes are decoupled, the spectral density $S_y(\nu)$ of the displacement $y(x,t)$ at a point x will be [3]

$$S_y(\nu) = |\alpha_{xx_i}|^2 S_{F_i}(\nu) \tag{15}$$

If the random forces $F_r(t)$, $F_s(t)$ are acting on the structure at the points x_r and x_s, respectively, we get [6]

$$S_y(\nu) = \alpha^*_{xx_r}\alpha_{xx_r} S_{F_r}(\nu) + \alpha^*_{xx_r}\alpha_{xx_s} S_{F_r F_s}(\nu) + \alpha_{xx_s}\alpha^*_{xx_r} S_{F_s F_r}(\nu) + \alpha^*_{xx_s}\alpha_{xx_s} S_{F_s}(\nu) \tag{16}$$

where $F_{rs}(\nu)$, $F_{sr}(\nu)$ are intercorrelation spectral densities and α^* is the harmonic conjugate of α. For the action of the random /3

forces $F_1, F_2, \ldots F_k$ at the levels $x_1, x_2, x_3, \ldots x_k$ we get the generalized form

$$S_y(\nu) = \sum_{r=1}^{k} \sum_{s=1}^{k} \alpha_{xx_r} \alpha_{xx_s} S_{F_r F_s}(\nu) \tag{17}$$

The intercorrelation effect can be easily brought to light for different limiting cases. For the random forces F_r and F_s that come from independent sources $F_{rs}(\nu) = F_{sr}(\nu) = 0$, while for directly correlated sources, since c is a constant, we get $F_r(t) = CF$ and as a result

$$S_{F_r F_s}(\nu) = S_{F_s F_r}(\nu) = c F_s(\nu); \quad S_{F_r}(\nu) = c^2 S_{F_s}(\nu) \tag{18}$$

Theoretically, weaving looms operate at the same frequency. Practically, however, there exist slight frequency variations around a central value, which leads to a phase difference similar to random variation. As a result, the aggregate of individual vibration sources at a level can be regarded as a narrow-band random force. Recordings of the vibratory displacements and accelerations in weaving mills (shown in Fig. 2) confirm the assertion that random forces are also of this type

Fig. 2.

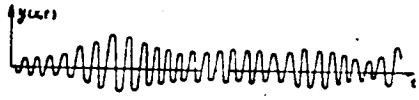

Fig. 3.

Because narrow-band random excitation can be nicely approximated by excitation of the narrow-band white noise type [5] (Fig. 3), assuming a constant spectral density $S_{F_i}(\nu) = ct$ along the entire band, the mean quadratic value of the displacement is [3]

$$\langle y^2 \rangle = S \int_0^{\infty} |\alpha_{xx_i}(\nu)|^2 d\nu \tag{19}$$

the response function α_{xx_i} being defined by the series $\Sigma \alpha_{xx_i}^{(n)}$. In the case of weak damping, the system will receive energy only in the neighborhood of the proper frequencies, and as a result, we can assume that the value of the total response can be approximated by the sum of the responses associated with each normal vibration mode. Taking into account the equation for α_{xx_i}, Eq. (19) can be written

$$\langle y^2 \rangle = S \sum_{n=1}^{\infty} \int_0^{\infty} |\alpha_{xx_i}^{(n)}(\nu)| d\nu = \sum_{n=1}^{\infty} \langle y_n^2 \rangle \tag{20}$$

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If we further take into account that the spectral density of the random force is of the narrow-band white noise type (Fig. 2), then

$$S_y(\nu) = \begin{cases} |\alpha_{n,m}|^2 S & \text{on } \nu \in [\nu_a, \nu_b] \\ 0 & \end{cases}$$

and after calculations Eq. (20) becomes

$$\langle y^2 \rangle = S \sum_{\omega_a < \omega_n < \omega_b} \frac{\chi_n^2(x) \chi_n^2(x_i)}{M_n} \cdot \frac{\pi}{2} \cdot \frac{1}{S_n \omega_n^2} \left[I\left(\frac{\omega_b}{\omega_n}, S_n\right) - I\left(\frac{\omega_a}{\omega_n}, S_n\right) \right] \tag{21}$$

with $\omega_n \in [\omega_a, \omega_b]$. The factor $I\left(\frac{\omega}{\omega_n}, \omega_n\right)$ is given [4] by the expression

$$I(\Omega, S_n) = \frac{1}{\pi} \operatorname{arctg} \frac{2 S_n (\Omega)}{1 - (\omega/\omega_n)^2} + \frac{\pi_2}{4\pi\sqrt{1 - S_n^2}} \ln \frac{1 + (\nu/\nu_{on})^2 + 2\sqrt{1 - S_n^2}(\nu/\nu_{on})^2}{1 + (\nu/\nu_{on})^2 - 2\sqrt{1 - S_n^2}(\nu/\nu_{on})} \tag{22}$$

It is demonstrated that there exists in the equation

$$I\left(\frac{\omega_b}{\omega_n}, S_n\right) - I\left(\frac{\omega_a}{\omega_n}, S_n\right) \leq 1 \tag{23}$$

This difference in the functions I is close to unity when the band width $\omega_b - \omega_a$ that includes the resonant frequency of the proper mode is wide as compared with the width of the resonant bands $\Delta\omega$ of the proper mode (Fig. 4).

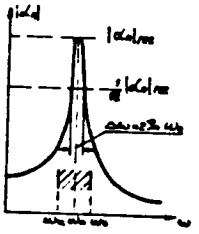

Fig. 4.

In the case of white noise, the difference has its maximum value because

$$I(\omega_n, S_n) - I(0, S_n) = 1 \tag{24}$$

For weaving looms, the narrow band could coincide with the resonant frequency $\Delta\omega$ for a single vibration mode, and for the other proper vibration modes the above-mentioned distance is practically zero. In this case, owing to the action of "k" on correlated forces, the mean quadratic value of the displacement will be

$$\langle y^2 \rangle = \frac{\pi}{2} S \frac{\chi_n^2(x)}{S_n \omega_n^2 M_n} \left[I\left(\frac{\omega_b}{\omega_n}, S_n\right) - I\left(\frac{\omega_a}{\omega_n}, S_n\right) \right] \sum_{i=1}^{k} \chi_n^2(x_i) \tag{25}$$

4. Conclusions

The method described in the present paper permits study of the effect of classical weaving looms on storied buildings. /4

Utilization of the calculation formulas as early as the designing stage is predicated on study of the behavior of an equivalent model embodied on a reduced scale, the elastic or damping characteristics being determined experimentally [4].

The range of applicability of the equations obtained can also be extended to the case of powerful damping, if the uncoupled equations that approximate the structure's vibrations are written, as was done in Ref. [7].

Summary

The authors derive expressions for the displacements of a many storied building subjected to the action of classical weaving looms located at different levels of the building. The building was regarded as a vertical fixed beam with a uniformly distributed mass as well as concentrated masses at each level. /4

The calculation relations were derived on the assumption of harmonic variation of the forces acting at each level as well as the assumption of narrow-band stationary random excitatory forces.

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INSULATION OF NONLINEAR AND RANDOM VIBRATIONS IN THE MINING INDUSTRY

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The vibrations of some mining implements (rotary crushers, impact mills, resonance screens, Airmix and Rotoclone fans) have a nonlinear character because the differential equations with partial derivatives that describe the dynamic processes of these implements from the mathematical point of view contain nonlinear terms. In general, these implements are gripped at their base by means of elastic elements made of rubber. Well-known is the fact that this material (rubber) has a nonlinear characteristic. Moreover, the effort-deformation curve in the case of stress incident to compression of the rubber is influenced by the shape of the insulator (defined as the ratio between loaded surface and free surface), so that in the elastic law of rubber insulators higher powers of the deformation appear. Thus, the above-mentioned mining implements, together with their elastic supports, constitute aggregates whose dynamic behavior is nonlinear. /4

The perturbing forces and couples transmitted by rotary crushers, impact mills and other mining implements have a deterministic, harmonic character owing to rotating parts. Superposed on these harmonic excitations, however, are random excitations caused by the crushing and sorting of lumps of ore that have random dimensions; these random excitations are randomly distributed in the machines, have random physical-mechanical characteristics and crop up in the working parts of the implements at random intervals of time.

In Rumania, in calculating and designing the insulation of mining implements, the nonlinear behavior of the implement-base aggregate or the random character of the excitations has not been taken into account, which, in some cases, has led to choosing inadequate solutions. /4

The coefficient of elasticity of a rubber insulator stressed incident to compression can be expressed by the relation:

$$k = k_0(1+\gamma x^2) \qquad (1)$$

with $\gamma > 0$. Eq. (1) represents a hard symmetric spring.

The differential equation of the vibrations of the implements gripped at their base by means of such an insulator is:

$$m\ddot{x} + c\dot{x} + k_0 x + k_0 \gamma x^3 = Q \sin \omega t \qquad (2)$$

The three vibrations of this system must of necessity contain a harmonic of the third order, of sufficiently great amplitude to produce an acceleration that increases in both directions (Fig. 1).

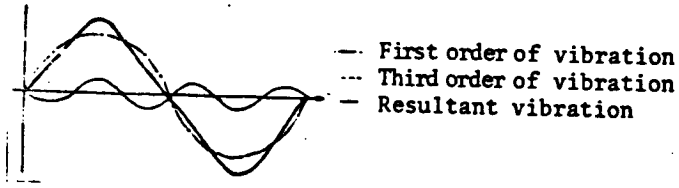

Fig. 1.

The phenomenon of frequency reduction characteristic of nonlinear systems crops up. As a result, the subharmonic oscillations of the order 1/3 can be generated when they are in resonance with the proper frequency (which varies with the amplitude since the system is nonlinear). For judicious calculation of vibration insulators the proper frequencies and the conditions for the appearance of the subharmonic oscillations of the order 1/3 must be determined.

Let us introduce the following adimensional coefficients:

-- the relative damping: $\varepsilon = \frac{c}{m\omega_0}$

-- the relative frequency: $\theta = \frac{\omega}{\omega_0}$ (3)

-- the relative pulse: $\gamma = \frac{Q}{k_0}$

($\omega_0 = \sqrt{k_0/m}$ is the natural frequency of the linearized system). /4

Taking a new variable $\tau = \omega_0 t$ and introducing the coefficients of Eq. (3), Eq. (2) becomes:

$$x'' + \varepsilon x' + x + \gamma x^3 = \varepsilon \cos \theta\tau \qquad (4)$$

where:

$$x' = \frac{dx}{d\tau} \; ; \; x'' = \frac{d^2x}{d\tau^2}$$

The equation of the undamped free vibrations are obtained from Eq. (4) by setting $q = 0$; $\rho = 0$:

$$x'' + x + \gamma x^3 = 0 \qquad (5)$$

In the solution of Eq. (5) there exists no constant term and in a first approximation we have:

$$x = a \sin \xi\tau \tag{6}$$

Introducing the solution of Eq. (6) into Eq. (5) and identifying the terms in sin $\xi\tau$, we get:

$$\xi^2 = 1 + \frac{1}{4}\gamma a^2 \tag{7}$$

Equation (7) represents the amplitude-frequency characteristic in a first approximation for the nonlinear system of Eq. (5). It can be seen that the frequency increases with the amplitude.

For the second approximation we take:

$$x = a \sin \xi\tau + b \sin 3\xi\tau \tag{8}$$

Writing that the solution of Eq. (8) satisfies Eq. (5), it follows from identification of the terms in sin $3\xi\tau$ that:

$$b - 9b\xi^2 - \frac{\gamma}{4}a^3 + 3\gamma a^2 b = 0$$

Introducing into this equation the value of ξ given by Eq. (7) and taking into account that $\gamma a^2 \ll 1$, we get:

$$b = -\frac{\gamma a^3}{32} \tag{9}$$

It was shown in reference [1] that for subharmonic oscillations of the order 1/3 to appear, it is necessary and sufficient for the mechanical work furnished by the pulse and absorbed by the harmonic of the third order when it is in resonance with the pulse of the same frequency to be greater than the mechanical work absorbed by the damping of the total vibration (assuming that this second mechanical work is approximately equal to that given only by the component of the first order of the total vibration).

The mechanical work furnished by the pulse of the third order (n = 3): g sin $3\xi\tau$, for three oscillations is: /4

$$L_a = -3\pi g \lambda_0 \tau \tag{10}$$

The mechanical work absorbed by the damping $T = \rho x'$ for an oscillation of the component of the first order (x_I = a sin $\xi\tau$) has the value:

$$L_r = \rho \lambda_0 a^2 \tau \tag{11}$$

The condition for the appearance of subharmonic oscillations of the order 1/3 ($L_{III} > L_I$), taking into account Eqs. (10) and (11), is:

$$\rho < \frac{\delta}{12}\eta^2 \qquad (12)$$

For the specified implements of the mining industry (which means for most practical cases) Eq. (12) is satisfied so that the priming of the subharmonic vibrations of the order 1/3 cannot be impeded and in calculating rubber insulators the subharmonic resonance must be taken into account. In this case, the implements will be protected for 1/3 of the perturbing frequency by calculating the forces of inertia for this frequency, the amplitudes being determined from Eqs. (7) and (9).

Subharmonic vibrations of the order 1/3 were detected experimentally in hammer mills for coal and in Airmix and Rotoclone ventilators in briquetting works [2]. For these implements we designed insulators of the type shown in Fig. 2, made of rubber with a hardness of 50 Shore and $E = 25 \cdot 10^5 N/m^2$, $\sigma_a = \varepsilon \cdot \omega^2 \frac{\rho}{\beta_1}$, $\varepsilon = 4\%$, with metallic inertias of stainless steel. With these insulators we obtained coefficients of the transmissability with values below 10% for the subharmonic frequency.

The random vibrations encountered in mining implements on which experimental investigations were conducted (rotary crushers, resonance screens) can be represented by a Gaussian normal distribution function with the properties of such a distribution (the vibrations are stationary and ergodic, and the response is also a Gaussian vibration).

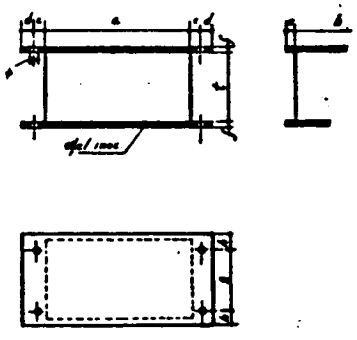

Fig. 2. 1. Stainless steel.

The excitation produced by rotary crushers (Symons) is a random vibration with a relatively wide frequency spectrum but with a spectral power density concentrated in the 8-13 Hz band. The response of the base is a random vibration with a narrow frequency spectrum in the 38-62 Hz band. /4

The insulation of these implements by means of rubber insulators was calculated on the basis of the spectral power density. It is known [3] that between the spectral power density of the input $W_e(\omega)$ and that of the output $W_T(\omega)$ there exists the following relation:

$$W_a(\omega) = W_e(\omega)|A(i\omega)|^2 \qquad (13)$$

in which $A(i\omega)$ is the response function for a harmonic excitation (the response function $A(i\omega)$ is the Fourier transform of the response pulse function $h(t)$, i.e., of the system's response to the unit pulse $\delta(t)$: $A(i\omega) = \int_0^\infty e^{i\omega\tau} h(\tau)\,d\tau$; the function $A(i\omega)$ is called the transfer function).

In view of the type of distribution and the fact that the response of the base has a narrow frequency spectrum, the function $A(i\omega)$ becomes for Symons crushers the absolute transmissability for the insulation system in the case of the sinusoidal vibration T_A, so that Eq. (13) assumes the form:

$$W_f(\omega) = W_m(\omega)\, T_A^2 \qquad (14)$$

in which $W_f(\omega)$ is the mean quadratic density of the acceleration of the base. /4

Using the high experimental values, from Eq. (14) we can determine the necessary value of the absolute transmissibility T_A of the insulator system for the insulation of the entire excitation spectrum. Once the value of T_A has been determined, the insulators are calculated as in the case of sinusoidal excitations. An important factor in the insulation of random vibrations is insulator deformation. In the case of Symons crushers, taking into account that the response is a narrow-band vibration, the mean quadratic value of the response is given by the relation [3]:

$$\overline{x^2} = \frac{\pi}{4\xi} \cdot \frac{W_n(\omega_n)}{\omega_n^3} \qquad (15)$$

in which ξ is the relative damping ($\xi = c/c_c$, c_c being the critical damping), and ω_n is the resonance pulsation.

Thus, the necessary deformation of rubber insulators for random vibrations can be calculated from Eq. (15) by substituting

$$\omega_n = 2\pi f_n, \quad W_a = W_f = W_m T_{A\max}^2, \quad T_{A\max} = \delta = \frac{1}{2\pi^2} \sqrt{\frac{W_m T_{A\max}}{f_n^3}} \qquad (16)$$

Also determined in the experimental studies of references [2, 4, 5] were the values of the mean square deviations σ of the displacements for the principal types of Symons crushers used in

Romania, so that it was possible to determine the impact probability of the insulators at the stop, which turned out to be at 100 cycles for its value calculated on the basis of Eq. (16) and probably at 10^6 cycles if we take for the deformation a value three times greater than that resulting from Eq. (16).

Summary

The authors calculate the insulation of the nonlinear and random vibrations of some ore preparing and sorting implements: rotary crushers, resonance screens, hammer mills, etc. /50

The appearance of subharmonic vibrations is analyzed, and the conditions for their appearance are determined. A method is given for calculating the insulation of these vibrations by means of elastic elements made of rubber.

The insulation of the random vibrations produced by Symons crushers is calculated by determining the transmissability and deformation of the insulation system for a narrow-band random response.

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RANDOM VIBRATIONS OF QUADRATIC DAMPING SYSTEMS

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1. Exact Solutions of the Fokker-Planck Equation

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Exact solutions of the Fokker-Planck equation associated with nonlinear systems of the order two excited by white noise have been determined only for the stationary probability density of equations of the form [1]

$$\ddot{x} + f(H)\dot{x} + g(x) = N(t), \quad x(0) = x_0, \ \dot{x}(0) = \dot{x}_0, \quad E[N(t)] = 0, \ E[N(t)N(t+\tau)] = 2D\delta(\tau), \tag{1.1}$$

where

$$H(x,\dot{x}) = \frac{1}{2}\dot{x}^2 + \int_0^x g(\eta)\,d\eta. \tag{1.2}$$

The Fokker Planck equation associated with the system of Eq. (1.1) is

$$\frac{\partial p}{\partial t} = -\dot{x}\frac{\partial p}{\partial x} + \frac{\partial}{\partial \dot{x}}[(g(x) + f(H)\dot{x})p] + D\frac{\partial^2 p}{\partial \dot{x}^2}, \quad p(x,\dot{x};0|x_0,\dot{x}_0) = \delta(x-x_0)\delta(\dot{x}-\dot{x}_0). \tag{1.3}$$

If there exists a stationary probability density $p_s(x,\dot{x})$ toward which $p(x,\dot{x},t)$ tends when $t \to \infty$, then it satisfies the stationary Fokker-Planck equation

$$g(x)\frac{\partial p_s}{\partial \dot{x}} - \dot{x}\frac{\partial p_s}{\partial x} + \frac{\partial}{\partial \dot{x}}\left[f(H)\dot{x} + D\frac{\partial p_s}{\partial \dot{x}}\right]. \tag{1.4}$$

In order for a solution $p_s(x,\dot{x})$ of this equation to represent a probability density, it must satisfy the conditions

$$p_s(x,\dot{x}) \geq 0, \tag{1.5}$$

$$\int_{-\infty}^{\infty}\int_{-\infty}^{\infty} p(x,\dot{x})\,dx\,d\dot{x} = 1. \tag{1.6}$$

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The limiting conditions for Eq. (1.4) are

$$p_s(x,\dot{x}) \to 0 \quad \text{for} \quad x^2+\dot{x}^2 \to \infty, \tag{1.7}$$

and as a result $\frac{\partial p_s}{\partial x} \to 0$, $\frac{\partial p_s}{\partial \dot{x}} \to 0$ for $\ldots$.

The existence and uniqueness of solutions of the Fokker-Planck equation and the convergence towards the stationary state was recently investigated by Kushner [2].

Eq. (1.4) can be immediately solved if the following separation is utilized:

$$g(x)\frac{\partial p_s}{\partial \dot{x}} - \dot{x}\frac{\partial p_s}{\partial x} = 0,$$
$$\frac{\partial}{\partial \dot{x}}\left[f(H)\dot{x} + b\frac{\partial p_s}{\partial \dot{x}}\right] = 0. \tag{1.8}$$

The particular solution of this equation that satisfies the conditions of Eqs. (1.5), (1.6) and (1.7) is given by

$$p_s(x,\dot{x}) = \frac{\exp\left[-\frac{1}{b}\int_0^H f(\eta)\,d\eta\right]}{\int\int \exp\left[-\frac{1}{b}\int_0^H f(\eta)\,d\eta\right]\,dx\,d\dot{x}} \tag{1.9}$$

2. The Equivalent Nonlinear System

Let us consider the equation

$$\ddot{x} + 2\zeta_1\omega_1\dot{x} + \omega_1^2 x = N(t). \tag{2.1}$$

We replace this equation by an equation of the form of Eq. (1.1), namely:

$$\ddot{x} + 2\zeta_e\dot{x}\sqrt{2H} + \omega_1^2 x = N(t), \tag{2.2}$$

so that the mean square error

$$\ldots = 4E[\ldots \tag{2.3}$$

Applying the minimization method, we get /5

$$k_e = \frac{E[\dot{x}^2|\dot{x}|\sqrt{2H}]}{E[\dot{x}^4] + \omega_n^2 E[\dot{x}^2 x^2]} k. \tag{2.4}$$

In this equation the mean operator E is defined in relation to the stationary probability density $p_S(x,\dot{x})$ corresponding to that of Eq. (2.2), i.e.,

$$E[U(x,\dot{x})] = \int_{-\infty}^{\infty}\int_{-\infty}^{\infty} U(x,\dot{x}) p_S(x,\dot{x}) dx d\dot{x}. \tag{2.5}$$

Putting into Eq. (1.9)

$$f(H) = 2k_e\sqrt{2H}, \quad g(x) = \omega_n^2 x, \tag{2.6}$$

and integrating, we get

$$p_S(x,\dot{x}) = \frac{3\omega_n}{2\pi\Gamma(1/3)}\left(\frac{2k_e}{3b}\right)^{1/3} \exp\left[-\frac{2k_e}{3b}\left(\dot{x}^2+\omega_n^2x^2\right)^{3/2}\right] \tag{2.7}$$

The equivalent damping coefficient of Eq. (2.4) corresponding to this probability density is given by

$$k_e = \frac{8}{3\pi} k. \tag{2.8}$$

Taking into account the last equation, the probability density of Eq. (2.7) is given by the following expression

$$p_S(x,\dot{x}) = \frac{3\omega_n}{2\pi\Gamma(1/3)}\left(\frac{16k}{9\pi b}\right)^{1/3} \exp\left[-\frac{16k}{9\pi b}\left(\dot{x}^2+\omega_n^2x^2\right)^{3/2}\right]. \tag{2.9}$$

3. Mean Square Response and Optimization of Damping

Of particular interest in practical applications are the relative displacement $x(t)$, the relative speed $\dot{x}(t)$ and the absolute acceleration $\ddot{z}(t) = \ddot{x}(t) + N(t)$. In general, the mean values of these random processes, as well as their mean square values, are approximated by analogous values for the equivalent system.

The mean square values $E[x^2]$ and $E[\dot{x}^2]$ calculated in relation to the probability density of Eq. (2.9) are of the form:

$$E[x^2] = \frac{1}{2}\left(\frac{3\sqrt{\pi}}{4}\right)^{4/3}\frac{\Gamma(4/3)}{\Gamma(2/3)}\frac{1}{\omega_n^2}\left(\frac{D}{\kappa}\right)^{2/3} \qquad (3.1)$$

$$E[\dot{x}^2] = \frac{1}{2}\left(\frac{3\sqrt{\pi}}{4}\right)^{4/3}\frac{\Gamma(4/3)}{\Gamma(2/3)}\left(\frac{D}{\kappa}\right)^{1} \qquad (3.2)$$

In order to calculate the absolute mean square acceleration $\ddot{z}(t)$, which is essential in optimizing the system's characteristic, we use our initial equation, Eq. (2.1). Thus,

$$E[\ddot{z}^2] = E[(2\kappa|\dot{x}|\dot{x} + \omega_n^2 x)^2] = 4\kappa^2 E[\dot{x}^4] + 4\kappa\omega_n^2 E[|\dot{x}|\dot{x}x] + \omega_n^4 E[x^2], \qquad (3.3)$$

and, therefore,

$$E[\ddot{z}^2] = \left(\frac{\pi}{4}\right)^{1/3}\left[3\left(\frac{\pi}{4}\right)^{1/3} D^2\left(\frac{D}{\kappa}\right)^{-2/3} + \frac{\omega_n^2}{2}\left(\frac{D}{\kappa}\right)^{2/3}\right]. \qquad (3.4)$$

The optimum damping coefficient k_0 that minimizes this expression is given by the solution of the following equation:

$$\frac{\partial E[\ddot{z}^2]}{\partial \kappa} = 0. \qquad (3.5)$$

Solving this equation, we get

$$k_0 = \frac{4}{3\sqrt{\pi}}\left(\frac{\Gamma(4/3)}{3}\right)^{3/4}\left(\frac{\omega_n^2}{D}\right)^{1/2}. \qquad (3.6)$$

4. Optimization of Automobile Suspensions

A number of studies [3, 4, 5] dealing with the statistical description of different types of roads have shown that the spectral densities of the random processes that model the longitudinal profile of roads can be approximated by expressions of the form

$$S(\Omega) = \qquad (4.1)$$

where

$$\Omega = \frac{1}{\lambda} \qquad (4.2)$$

λ being the wavelength. The distribution of the road roughness amplitude is assumed to be Gaussian [6].

If an automobile is traveling at a constant velocity v, the spectral density of the excitation caused by road roughness can be obtained from Eqs. (4.1) and (4.2) by setting /5

$$\lambda = \frac{v}{f} , \tag{4.3}$$

f being the time frequency. Denoting by $S_v(\omega)$ the spectral density of the excitation for velocity v, $\omega = 2\pi f$ being the circular frequency, we get

$$S_v(\omega) = C v^N \omega^{-N} . \tag{4.4}$$

If now our model of the front or rear of an automobile is an oscillating system with a single degree of liberty having a linear elastic characteristic and a quadratic damping characteristic, the equation of motion is one and the same as Eq. (2.1) if N = 4. Indeed, if $h_v(t)$ is the random process having the spectral density $S_v(\omega)$ given by Eq. (4.4), then

$$\dot{h}_v(t) = N_v(t) ,$$
$$E[N_v(t)] = 0 , \; E[N_v(t) N_v(t+\tau)] \tag{4.5}$$

The absolute mean square acceleration of the suspended mass of an automobile is generally considered to be one of the most significant parameters in estimating comfort [6, 7, 8, 9]. Taking into account Eqs. (3.6) and (4.5), the optimum damping coefficient k_0 is given by

$$k_0 = \frac{4}{\pi} \left(\frac{\Gamma(4/3)}{3} \right)^{3/4} \left(\frac{\omega^2}{C} \right)^{1/2} \frac{1}{v^2} . \tag{4.6}$$

As a result, optimum damping is less for greater velocities, and so is parameter C, which is a measure of road roughness. On the other hand, if the suspension's natural frequency ω_n increases, then so does the optimum damping. Practically speaking, Eq. (4.6) defines one of the most important conditions for running automobiles and optimizing the damping for these conditions, namely: the kind of road, the driving and loading speed, the latter influencing the suspension's local proper oscillation frequency. The above results were confirmed qualitatively by way of experiment by measuring the mean square acceleration of the suspended mass in the case of an automobile equipped with adjustable shock absorbers [10].

Summary

An oscillating system with quadratic damping subjected to white noise excitation is replaced by a nonlinear, statistically equivalent system for which the associated Fokker-Planck equation can be exactly solved. The mean square responses are calculated and the optimum damping coefficient is determined with respect to the minimum mean square acceleration criteria. An application of these results to the optimization of automible suspension damping is given.

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RANDOM VIBRATIONS OF LINEAR VISCOELASTIC BEAMS WITH LUMPED MASSES

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The calculation method that is going to be presented constitutes a manner of controlling the vibration level of a beam subjected to random excitation by attaching lumped masses along its length. /5

1. The Equation of Motion

Let us consider a linear beam of length ℓ, with a constant section, homogeneous and isotropic, made of a viscoelastic material with a constituent law of the form

$$P(\sigma) = Q(\varepsilon) \ , \tag{1.1}$$

where:

$$P = \sum_{k=0}^{p} p_k \frac{\partial^k}{\partial t^k} \ , \quad Q = \sum_{k=0}^{p} q_k \frac{\partial^k}{\partial t^k} \ , \tag{1.2}$$

where p_k and q_k are constants.

Let us assimilate the beam to an orthogonal system xOu with its origin at one of the ends.

Let us assume that along the bar, at the points $M_k(x_k, 0)$ the masses $m_k (k = 1, \ldots, n)$ are attached and that the beam is acted upon at every point by a transversal force F(t), random in time, stationary, almost certainly derivable p times, with a rational spectral density.

The equation of motion is [2]

$$\rho A Q\left(\frac{\partial^2 u}{\partial t^2}\right) + \left[\rho A + \sum_{k=1}^{n} m_k \delta(x - x_k)\right] P\left(\frac{\partial^2 u}{\partial t^2}\right) = P[F(t)] \tag{1.3}$$

where ρ is the specific mass, and A is the beam's sectional area.

We shall consider limiting conditions of the form:

I. The beam is simply resting at both ends:

$$u(0,t) = u(l,t) = \frac{\partial^2 u}{\partial x^2}\Big|_{(0,t)} = 0 \qquad (1.4)$$ /60

II. The beam is fixed at both ends:

$$u(0,t) = u(l,t) = \frac{\partial u}{\partial x}\Big|_{(0,t)} = 0 \qquad (1.5)$$

III. The beam is fixed at one end and simply resting at the other:

$$u(0,t) = u(l,t) = \frac{\partial u}{\partial x}\Big|_{(0,t)} = 0 \qquad (1.6)$$

IV. The beam is fixed at one end and free at the other:

$$u(0,t) = \frac{\partial u}{\partial x}\Big|_{(0,t)} = \frac{\partial^2 u}{\partial x^2}\Big|_{(l,t)} = \frac{\partial^3 u}{\partial x^3}\Big|_{(l,t)} = 0. \qquad (1.7)$$

Inasmuch as we shall have in view a uniformly asymptotic solution, which is, on the whole, stable, we shall impose no initial conditions on the mean square of Eq. (1.3).

2. Proper Vibration Forms

In order to determine the proper vibration forms, we successively impose on Eq. (1.3), considered to be homogeneous, the limiting conditions of Eqs. (1.4) ... (1.7).

Trying a solution of the form

$$u(x,t) = X(x)T(t), \qquad (2.1)$$

we get for a determination of X(x) the ordinary differential equation

$$\frac{\frac{\partial^4 X}{\partial x^4}}{\left[\rho A + \sum_{k=1}^{n} m_k \delta(x - x_k)\right] X} = \lambda^4. \qquad (2.2)$$

Using the limiting conditions of Eqs. (1.4) ... (1.7) for x = 0, we get

$$X_I(x) = \frac{X'(0)}{\lambda} T(\lambda x) + \frac{X''(0)}{\lambda^2} V(\lambda x) + \\ + \lambda \sum_{k=1}^{n} \mu_k X(x_k) V[\lambda(x-x_k)] H(x-x_k),$$

$$X_{II}(x) = \frac{X'(0)}{\lambda^2} U(\lambda x) + \frac{X'''(0)}{\lambda^3} V(\lambda x) + \\ + \lambda \sum_{k=1}^{n} \mu_k X(x_k) V[\lambda(x-x_k)] H(x-x_k), \qquad (2.3)$$

$$X_{III}(x) = X_{\overline{III}}(x) = X_I(x),$$

where $\mu_k = m_k/sA$, S, T, U, V are functions known from the deterministic theory of beam flexure and H is the Heaviside function.

Taking into account the limiting conditions of Eqs. (1.4) ... (1.7) for x = 1, we get

$$X_I(x) = \lambda \sum_{k=1}^{n} \mu_k X(x_k) \{T(\lambda x) \varphi_I(\lambda, l, x_k) + V(\lambda x) \psi_I(\lambda, l, x_k) + \\ + V[\lambda(x-x_k)] H(x-x_k)\},$$

$$X_{II}(x) = \lambda \sum_{k=1}^{n} \mu_k X(x_k) \{U(\lambda x) \varphi_{II}(\lambda, l, x_k) + V(\lambda x) \psi_{II}(\lambda, l, x_k) + \\ + V[\lambda(x-x_k)] H(x-x_k)\}, \qquad (2.4)$$

$$X_{III}(x) = \lambda \sum_{k=1}^{n} \mu_k X(x_k) \{U(\lambda x) \varphi_{III}(\lambda, l, x_k) + V(\lambda x) \psi_{III}(\lambda, l, x_k) + \\ + V[\lambda(x-x_k)] H(x-x_k)\},$$

$$X_{\overline{III}}(x) = \lambda \sum_{k=1}^{n} \mu_k X(x_k) \{U(\lambda x) \varphi_{\overline{III}}(\lambda, l, x_k) + V(\lambda x) \psi_{\overline{III}}(\lambda, l, x_k) + \\ + V[\lambda(x-x_k)] H(x-x_k)\},$$

where

$$\varphi_I = \frac{V(\lambda l) T[\lambda(l-x_k)] - T(\lambda l) V[\lambda(l-x_k)]}{T^2(\lambda l) - V^2(\lambda l)},$$

$$\varphi_{II} = \frac{V(\lambda l) U[\lambda(l-x_k)] - U(\lambda l) V[\lambda(l-x_k)]}{U^2(\lambda l) - V(\lambda l) T(\lambda l)},$$

$$\varphi_{III} = \frac{V(\lambda l) T[\lambda(l-x_k)] - T(\lambda l) V[\lambda(l-x_k)]}{U(\lambda l) T(\lambda l) - S(\lambda l) V(\lambda l)}, \qquad (2.5)$$

$$\varphi_{\overline{III}} = \frac{T(\lambda l) S[\lambda(l-x_k)] - S(\lambda l) T[\lambda(l-x_k)]}{S^2(\lambda l) - V(\lambda l) T(\lambda l)},$$

$$\psi_I = \frac{V(\lambda l) V[\lambda(x-x_k)] - T(\lambda l) T[\lambda(l-x_k)]}{T^2(\lambda l) - V^2(\lambda l)},$$

$$\psi_{II} = \frac{T(\lambda l) V[\lambda(x-x_k)] - U(\lambda l) U[\lambda(l-x_k)]}{U^2(\lambda l) - V(\lambda l) T(\lambda l)},$$

$$\psi_{III} = \frac{S(\lambda l) V[\lambda(x-x_k)] - U(\lambda l) T[\lambda(l-x_k)]}{U(\lambda l) T(\lambda l) - S(\lambda l) V(\lambda l)},$$

$$\psi_{\overline{III}} = \frac{V(\lambda l) T[\lambda(x-x_k)] - S(\lambda l) S[\lambda(l-x_k)]}{S^2(\lambda l) T(\lambda l) - V(\lambda l) T(\lambda l)}.$$

Into Eq. (2.4) we introduce $x = x_j$ ($j = 1, 2, \ldots n$) and for determination of the unknowns $X(x_k)$ we get four linear and homogeneous algebraic systems for the four cases under consideration. These can be given nontrivial solutions if

$$\det [\lambda \mu_n A_{nj} - \delta_{nj}] = 0 , \tag{2.6}$$

where

$$A_{nj} = \begin{cases} T(\lambda x_j)\Psi_{\mathrm{I}}(\lambda, l, x_n) + V(\lambda x_j)\Psi_{\mathrm{I}}'(\lambda, l, x_n) + V[\lambda(x_n - x_j)]H(x_n - x_j) \\ U(\lambda x_j)\Psi_{\mathrm{II}}(\lambda, l, x_n) + V(\lambda x_j)\Psi_{\mathrm{II}}'(\lambda, l, x_n) + V[\lambda(x_n - x_j)]H(x_n - x_j) \\ U(\lambda x_j)\Psi_{\mathrm{III}}(\lambda, l, x_n) + V(\lambda x_j)\Psi_{\mathrm{III}}'(\lambda, l, x_n) + V[\lambda(x_n - x_j)]H(x_n - x_j) \\ U(\lambda x_j)\Psi_{\mathrm{IV}}(\lambda, l, x_n) + V(\lambda x_j)\Psi_{\mathrm{IV}}'(\lambda, l, x_n) + V[\lambda(x_n - x_j)]H(x_n - x_j) \end{cases} \tag{2.7}$$

Equation (2.6) is a transcendental equation in λ. If $\lambda_\lambda, \ldots, \lambda_m, \ldots$ represents the roots of this equation, we have, correspondingly, the solutions $X_\lambda, \ldots, X_m, \ldots$ for each of the four cases. Any of the solutions determined in this manner is one of the proper vibration forms.

Inasmuch as Eqs. (2.2) that determine them are identical with those of the elastic case, the properties of the proper modes coincide in the two cases.

In particular, the property of orthogonality

$$\int_0^l X_n(x) X_m(x) [\rho A + \sum_{j=1}^{n} m_j \delta(x - x_j)]\, dx = N_n^2 \delta_{nm} , \tag{2.8}$$

is kept. /6

3. Solution of the Equation of Motion

We are seeking a solution of Eq. (1.3) in the form

$$u(x,t) = \sum_{n=1}^{\infty} X_n(x) T_n(t) , \tag{3.1}$$

where $X_k(x)$ are the proper functions and $T_k(t)$ are unknown random functions.

Setting

$$F(t) = F(t) \sum_{n=1}^{\infty} a_n X_n , \tag{3.2}$$

introducing Eqs. (3.1) and (3.2) into Eq. (1.3) and taking into account Eqs. (2.2) and (2.7), we get

$$P\left(\frac{d^2T_m}{dt^2}\right) + \lambda_m' IQ(T_m) = A_m P[F(t)], \quad m = 1, 2, \ldots, \tag{3.3}$$

where

$$A_m = -\frac{1}{N_m^2 gA} \sum_{k=1}^{\infty} \sum_{j=1}^{k} X_m(x_k) X_k(x_j). \tag{3.4}$$

The uniformly asymptotic, on the whole stable stationary solution is written in the mean square of Eq. (3.3) in the form

$$T_m(t) = \int_0^{\infty} h_m(\tau) F(t-\tau)\, d\tau, \tag{3.5}$$

where $h_m(\tau)$ is the transfer function of the unit pulse for the system of Eq. (3.3).

The frequency characteristic of the system of Eq. (3.3) is

$$H_m(i\omega) = A_m \frac{\sum_{j=0}^{p} p_j (i\omega)^j}{\sum_{j=0}^{r} \eta_{j,m} (i\omega)^j}, \tag{3.6}$$

where $r = \max(p + 2, q)$. The function $h_m(\tau)$ is given by the relation

$$h_m(\tau) = \frac{1}{2\pi} \int_{-\infty}^{\infty} H_m(i\omega) e^{i\omega\tau}\, d\omega. \tag{3.7}$$

The solution of Eq. (1.3) is /6.

$$u(x,t) = \sum_{k=1}^{\infty} X_k(x) \int_0^{\infty} h_k(\tau) F(t-\tau)\, d\tau. \tag{3.8}$$

The mean square of u(x,t) has the form

$$M_a^2(x) = \sum_{j=1}^{\infty}\sum_{k=1}^{\infty} X_j(x)X_k(x)\int_{-\infty}^{\infty}\int_{-\infty}^{\infty} h_j(\tau_1)h_k(\tau_2)k_F(\tau_1-\tau_2)\,d\tau_1 d\tau_2$$

$$= \sum_{j=1}^{\infty}\sum_{k=1}^{\infty} X_j(x)X_k(x)\int_{-\infty}^{\infty}\int_{-\infty}^{\infty} H_j(i\omega)H_k(-i\omega)S_F(\omega)\,d\omega \qquad (3.9)$$

where $k_F(\tau)$ and $S_F(\omega)$ are, respectively, the correlation function and the spectral density of the excitation F(t).

The integral of Eq. (3.9) is convergent inasmuch as $H_j(i\omega)$ are rational functions with a higher degree of the denominator than of the numerator and $S_F(\omega)$ is, by hypothesis, a rational function of the same type. It can be easily calculated with the theorem of residues.

By means of an adequate numerical analysis on the basis of the foregoing solution it is possible to determine the values of the controllable parameters (the magnitudes of the lumped masses and their distribution along the beam) that optimize the system in terms of a minimum mean square displacement.

Summary

In the paper there is presented a method of determining the mean square transversal deflection of an isotropic and homogeneous linear viscoelastic beam having a certain number of lengthwise distributed lumped masses. It is assumed that the beam is acted upon by a stationary random process uniformly distributed along the beam. The method is useful in vibration level control by means of additional lumped masses.

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THE RESPONSE OF MANY-STORIED STRUCTURES UNDER SEISMIC ACTIONS

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1. Introduction

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Theoretical study of structures under seismic actions has been undertaken (for example, in references [1, 2]) on the basis of simple mechanical models that take the place of the complex structures of many-storied buildings.

In what follows, we shall analyze the many-storied structure by assimilating it to a more complex mechanical model that consists of a vertical beam fixed at one end having a uniformly distributed mass p and on each floor concentrated masses $m_i (i = 1, 2, \ldots k)$. In treating the problem, we took into consideration two manners of erecting structures: on completely rigid ground and on ground with characteristics of linear deformability.

Since the effect of seismic action is characterized by the variation function of ground acceleration $\ddot{\Gamma}(t)$, we get calculation relations for the response, i.e., the amplitudes of any point of the structure, that are a function of the displacement $\Gamma(t)$.

Many studies undertaken to elucidate the laws of variation of seismic stresses, such as references [7, 8], permit specific statistical evaluations of certain regions.

Studies (for example, reference [7]) have shown some regularity, in mean values, of the intensity of seismic motion for a certain number of seisms, but also an almost constant distribution along a very wide frequency band.

Because every seismic motion consists of one or more consecutive motions having an irregular character, it is assumed that $\Gamma(t)$, the law that describes seismic motion, is a continuous function together with the first two derivatives, and its form depends on the results of seismic recordings in the zone under consideration.

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2. Differential Equation of the Equivalent Mechanical Model

The differential equation of the free transversal vibrations of a fixed beam with length ℓ and uniformly distributed mass p, to which an arbitrary number of concentrated masses $m_1, m_2, \ldots m_k$

are attached at the abscissa points x_1, x_2, ... x_k is [3, 4] (Fig. 1):

$$EI\frac{\partial^4 y}{\partial x^4} - \left[p - \sum_{i=1}^{k} m_i \delta(x - x_i)\right]\frac{\partial^2 y}{\partial x^2} = 0 \tag{1}$$

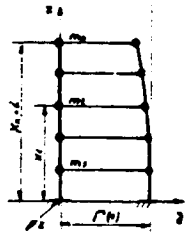

Fig. 1.

where δ is the Dirac distribution. The limiting conditions for the two above mentioned cases are:

-- completely rigid ground:

$$y(a,t) = \Gamma(t); \quad \frac{\partial^2 y(l,t)}{\partial x^2} = 0; \quad \frac{\partial^3 y(l,t)}{\partial x^3} = 0$$
$$\frac{\partial y(a,t)}{\partial x} = 0; \tag{2}$$

-- ground with characteristics of linear deformability (Fig. 2):

$$y(a,t) = \Gamma(t); \quad \frac{\partial^2 y(l,t)}{\partial x^2} = 0;$$
$$\frac{\partial y(0,t)}{\partial x} = \varphi(t); \quad \frac{\partial^3 y(l,t)}{\partial x^3} = 0; \tag{3}$$

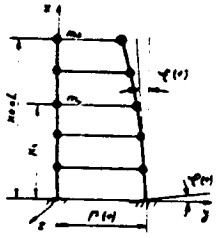

Fig. 2.

where $\phi(t)$ is a function of $\Gamma(t)$ through the medium of the ground's characteristics.

3. Solution of the Differential Equation for the Case of Completely Rigid Ground

Assuming a solution of the form:

$$y(x,t) = y(x)\, f(t) \tag{4}$$

we get the wave equation:

$$\frac{d^4 y}{dx^4} - \frac{\omega^2}{EI}\left[p + \sum_{i=1}^{k} m_i \delta(x - x_i)\right] y = 0 \tag{5}$$

The limiting conditions for Eq. (5) are given, depending on the case, by Eq. (2) or Eq. (3). /6

The solution of Eq. (5) is obtained by using the Laplace transform. After calculations we get:

$$y(x) = y(0)\, S(\lambda x) + \frac{1}{\lambda} y'(0) T(\lambda x) + \frac{1}{\lambda^2} y''(0) U(\lambda x) + \frac{1}{\lambda^3} y'''(0) V(\lambda x)$$
$$+ \frac{\omega^2}{EI\lambda^3} \sum_{i=1}^{k} m_i y(x_i) V(\lambda, x - x_i) H(x - x_i) \tag{6}$$

The notations are as follows: H is the Heaviside function;

$$y^{(i)}(0) = \frac{d^i y(x)}{dx^i}\Big|_{x=0}; \qquad \lambda = \sqrt{\frac{\omega^2 \rho}{EI}} \tag{7}$$

$$V(\lambda x) = \frac{\operatorname{sh}\lambda x - \sin\lambda x}{2}; \qquad U(\lambda x) = \frac{\operatorname{ch}\lambda x - \cos\lambda x}{2}$$

$$T(\lambda x) = \frac{\operatorname{sh}\lambda x + \sin\lambda x}{2} \qquad S(\lambda x) = \frac{\operatorname{ch}\lambda x + \cos\lambda x}{2} \tag{8}$$

Introducing the limiting conditions into Eq. (5), we get a system of equations, whence:

$$y^{(4)}(\ell, t) = \lambda^4 \left[\frac{A}{\rho} \sum_{i=1}^{k} m_i y(x_i, t) \varphi_2(\lambda, \ell, x_i) + \Gamma(t) \psi_2(\lambda, \ell)\right] \tag{9}$$

$$y^{(5)}(\ell, t) = \lambda^5 \left[\frac{A}{\rho} \sum_{i=1}^{k} m_i y(x_i, t) \varphi_3(\lambda, \ell, x_i) + \Gamma(t) \psi_3(\lambda, \ell)\right] \tag{10}$$

with the notations:

$$\varphi_2(\lambda, \ell, x_i) = 2\frac{S(\lambda, \ell - x_i) T(\lambda, \ell) - T(\lambda, \ell - x_i) S(\lambda, \ell)}{1 + \operatorname{ch}\lambda\ell \cdot \cos\lambda\ell};$$

$$\varphi_3(\lambda, \ell, x_i) = 2\frac{T(\lambda, \ell - x_i) V(\lambda, \ell) - S(\lambda, \ell - x_i) S(\lambda, \ell)}{1 + \operatorname{ch}\lambda\ell \cos\lambda\ell};$$

$$\psi_2(\lambda, \ell) = 2\frac{T^2(\lambda, \ell) - U(\lambda, \ell) S(\lambda, \ell)}{1 + \operatorname{ch}\lambda\ell \cdot \cos\lambda\ell}; \quad \psi_3(\lambda, \ell) = 2\frac{U(\lambda, \ell) V(\lambda, \ell) - S(\lambda, \ell) T(\lambda, \ell)}{1 + \operatorname{ch}\lambda\ell \cos\lambda\ell} \tag{11}$$

Substituting Eqs. (9) and (10) into Eq. (6) and taking into account the first two limiting conditions of Eq. (2), we get:

$$y(x, t) = \frac{1}{\rho} \sum_{i=1}^{k} m_i y(x_i, t) Y_i(x) + \Gamma(t) Z(x) \tag{12}$$

with the notations: /7

$$Y_i(x) = \varphi_2(\lambda, \ell, x_i) U(\lambda x) + \varphi_3(\lambda, \ell, x_i) V(\lambda x) + T_i(\lambda x - x_i) H(x - x_i)$$

$$Z(x) = \psi_2(\lambda, \ell) U(\lambda x) + \psi_3(\lambda, \ell) V(\lambda x) + S(\lambda x) \tag{13}$$

Setting $x = x_j$ ($j = 1, 2, \ldots k$), Eq. (12) is transformed into the system:

$$y(x_j, t) - \beta \sum_{i=1}^{k} m_i y(x_i, t) Y_{ij} = \Gamma(t) Z_j, \qquad j = 1, 2, \ldots k \tag{14}$$

with the notations:

$$\beta = \frac{A}{\rho}; \quad Y_i(x_j) = Y_{ij}; \quad Z(x_j) = Z_j \tag{14'}$$

From the system of Eq. (14) we get:

$$y(x_i,t) = \Gamma(t)\frac{1}{\Delta}\sum_{j=1}^{k}(-1)^{j+i+1}Z_j M_{ji} \qquad i=1,2,\dots,k \tag{15}$$

The equation of the proper pulsations is:

$$\Delta = \begin{vmatrix} \beta m_1\gamma_{11}-1 & \beta m_2\gamma_{12} & \cdots & \beta m_k\gamma_{1k} \\ \beta m_1\gamma_{21} & \beta m_2\gamma_{22}-1 & \cdots & \beta m_k\gamma_{2k} \\ \cdots & \cdots & \cdots & \cdots \\ \beta m_1\gamma_{k1} & \beta m_2\gamma_{k2} & \cdots & \beta m_k\gamma_{kk}-1 \end{vmatrix} \tag{16}$$

M_{ji} are the minors of the determinant Δ_i.

Substituting Eq. (15) into Eq. (12) and taking into account Eq. (4), we get:

$$y(x,t) = \left[Z(x) + \frac{\beta}{\Delta}\sum_{i=1}^{k} m_i\gamma_i(x)\sum_{j=1}^{k}(-1)^{j+i+1}Z_j M_{ji}\right]\Gamma(t) \tag{17}$$

Equation (17) represents the stationary component for the fundamental mode of the solution of differential equation (1).

4. Solution of the Differential Equation in the Case of Rotation Deformability

Let us consider that the deformable foundation ground permits the structure to rotate only about the oz axis. We assume that the deformability of the ground is perfectly elastic and reversible, as was shown in reference [1], if the condition $\sigma_t/C_\phi \leq 0.3$. The ratio σ_t/C_ϕ is called the ground's characteristic of deformability to seismic action, where σ_t is the effective pressure on the ground and C_ϕ is the coefficient of nonuniform contraction.

Taking into account the limiting conditions of Eq. (3), after a calculation analogous to the preceding, from Eq. (6) we get: /7

$$y(x,t) = \frac{\beta}{\rho}\sum_{i=1}^{k} m_i y(x_i,t)\gamma_i(x) + \Gamma(t)Z(x) + \Psi(t)W(x) \tag{18}$$

where we have written:

$$W(x) = \frac{1}{\lambda}\left[\Omega_2(\lambda,l)U(\lambda,x) + \Omega_3(\lambda,l)V(\lambda,x)\right]$$

$$\Omega_2 = 2\frac{T(\lambda,l)U(\lambda,l) - V(\lambda,l)S(\lambda,l)}{1+\mathrm{ch}\,\lambda l\cos\lambda l}\ ;\quad \Omega_3 = 2\frac{V^2(\lambda,l) - U(\lambda,l)S(\lambda,l)}{1+\mathrm{ch}\,\lambda l\cos\lambda l}\ ; \tag{19}$$

Setting $x = x_j$ (j = 1, 2, ... k), Eq. (18) is transformed into the system:

$$y(x_j,t) - \beta \sum_{i=1}^{k} m_i y(x_i,x_j) \gamma_{ij} - \Gamma(t) Z_j + \varphi(t) w_j \qquad j = 1, 2, \ldots k \tag{20}$$

with the notations of Eq. (14') and $W(x_j) = W_j$.

Making explicit $y(x_i t)$ (i = 1, 2, ... k) of the system of Eq. (20), we get:

$$y(x,t) = \Gamma(t)\left[Z(x) + \frac{\beta}{\Delta} \sum_{i=1}^{k} m_i \gamma_i(x) \sum_{j=1}^{k} (-1)^{i+j+1} Z_j M'_{ij}\right] + \varphi(t)\left[W(x) + \frac{\beta}{\Delta} \sum_{i=1}^{k} m_i \gamma_i(x) \sum_{j=1}^{k} (-1)^{i+j+1} W_j M'_{ij}\right] \tag{21}$$

5. Differential Equation of the Structure's Motion of Rotation

5.1. Case of Elastic Properties of the Foundation Ground

If κ is the ground's elastic constant, applying d'Alembert's principle, as can be seen from Fig. 3, we get:

$$(J_0 + \sum_{i=1}^{k} J_i)\ddot{\varphi}(t) - \sum_{i=1}^{k} m_i \ddot{y}(x_i,t) x_i - \int_0^l \rho \ddot{y}(x,t) x\, dx + \kappa\varphi = 0 \tag{22}$$

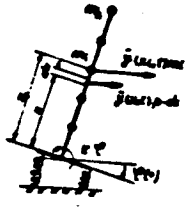

Fig. 3.

where J_0 denotes the moment of inertia with respect to the oz axis of the structure with a distributed mass and $J_i = m_i x_i^4$. Substituting $\ddot{y}(x,t)$, which results from Eq. (21), Eq. (22) becomes:

$$\ddot{\varphi}(t) + \nu^2 \varphi(t) = h\ddot{\Gamma}(t) \tag{23}$$

with the notations:

$$\nu^2 = \frac{y(x,t) = \Gamma(t)\alpha(x) + \varphi(t)\beta(x)}{J_0 + \sum_{i=1}^{k} J_i - \sum_{i=1}^{k} m_i \beta(x_i) x_i - \rho \int_0^l \beta(x) x\, dx} \tag{24}$$

$$h = \frac{\sum_{i=1}^{k} m_i \alpha(x_i) x_i + \rho \int_0^l \alpha(x) x\, dx}{J_0 + \sum_{i=1}^{k} J_i - \sum_{i=1}^{k} m_i \beta(x_i) x_i - \rho \int_0^l \beta(x) x\, dx} \tag{25}$$

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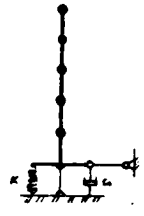
5.2. Case of Viscoelastic Properties of the Foundation Ground

If we consider ground with linear viscoelastic properties having the characteristics κ and C_t, we adopt an equivalent model of the Voigt type (Fig. 4). The differential equation in this case will be:

$$\ddot{\varphi}(t) + 2n\dot{\varphi}(t) + k^2\varphi(t) = h\ddot{\Gamma}(t) \tag{26}$$



with the notations of Eqs. (24), (25) and

$$n = \frac{C_t}{J_0 + \sum_{i=1}^{n} J_i - \sum_{i=1}^{n} m_i r(x_i) x_i - \rho \int_0^l r(x) x \, dx} \tag{27}$$

Fig. 4.

Differential equation (23) or (26) represents the connecting relation of the function $\phi(t)$ with the seismic acceleration function $\ddot{\Gamma}(t)$ through the medium of the elastic or viscoelastic properties of the foundation ground.

Conclusions

The fact that the distances between the levels of a building are equal, as well as the concentrated masses, simplifies calculations. Practically speaking, the structure will be studied by constructing a small-scale model. The calculation relations obtained will be used through the medium of simultaneity relations. The structure's elastic characteristics are determined by measuring the structure's response as a result of testing with harmonic or random narrow-band stresses with frequency modulation.

Summary

Supposing a seismic action which is characterized by the acceleration function of the seismic motion, the equations of motion in the direction of the seismic action have been determined. There have been analyzed the cases of completely rigid ground and of ground with linear deformability characteristics (rotation deformability). There has also been derived the equation of the rotation motion on the assumption of elastic and viscoelastic properties of the foundation ground. The differential equations have been written supposing a mechanical model of the many-storied structure as a fixed vertical beam with a uniformly distributed mass as well as concentrated masses on each floor.

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EXPERIMENTAL DETERMINATION OF THE VIBRATION LEVEL IN 5,100 kW ELECTRIC LOCOMOTIVE ASSEMBLIES

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The high vibration level that occurs during the operation of electric locomotives causes mechanical defects in the command mechanisms of auxiliary services. /75

This fact has generated the necessity of determining the magnitude of these vibrations in 5,100 kW locomotives.

While an electric locomotive was operating on the Craiova-Drobeta-Turnu Severin line, we conducted measurements on the following assemblies: unit assemblies S_4, S_5, S_6, S_7, ventilator T_{12} and compressor 36.1.

Determination Method

Inasmuch as it is a question of nonharmonic vibrations, it was necessary to measure all three parameters of motion -- displacement, speed and accleration -- as well as the frequency spectrum [2].

The magnitude of the displacement indicates whether the dynamic forces due to the machines can cause dangerous flexures in the supports of the apparatuses [1].

Knowledge of vibration speed is necessary in studying the transmission of noise through walls and other construction elements or in determining the radiation from plates with extensive surfaces.

Acceleration measurement is useful in calculating the powerful periodic forces that are one of the causes of defects in the apparatuses or of vibrations transmitted to the elements of other apparatuses, as well as in the event that powerful shocks take place [1].

Description of Devices Utilized

In order to measure the vibration parameters we utilized RFT-SM 231 and SM 241 measuring instruments with RD 35 a. seismic mass piezoelectric transducers. These permit measurement of the acceleration a, the velocity v and the vibration distance ξ in the 2 Hz - 15 Hz frequency range. /7

The instruments have an indicator unit on which the "slow and rapid" indicators, the effective value and the positive and negative peak value can be alternately read in dynamics in an exact manner from the dimensional point of view.

For the frequency analyses we included in the indicator components external OF 101 and OF 201 octave filters.

Comparative Determinations of the Assembly Vibrations from the Machine Room under Different Operating Conditions of Electric Locomotives

At different points of the above-mentioned assemblies in the case of a locomotive that was stationary, running individually or hauling freight trains and passenger trains on the Craiova-Drobeta-Turnu Severin line, we conducted measurements under the following running conditions: accelerated conditions with $I_{motor} > 0$, idling conditions with $I_{motor} = 0$ and braking conditions with $I_{motor} < 0$. A 380 t express train and a 1,344 t freight train were hauled.

The most important consideration from the point of view of resistance is to obtain proper equilibrium between the value of the unit effort that can be supported by the material and the value of the deformation that takes place [3]. For this reason, we placed a very great emphasis on knowing the value of the vibration amplitude ξ. The curves of Figs. 1, 2 and 3 show the variation in vibration amplitude determined in the assemblies of the locomotive's machine room.

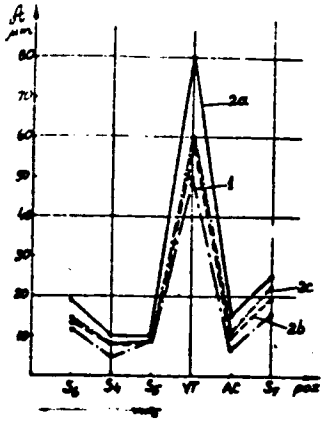

Fig. 1. Vibration amplitude curve for locomotive assemblies

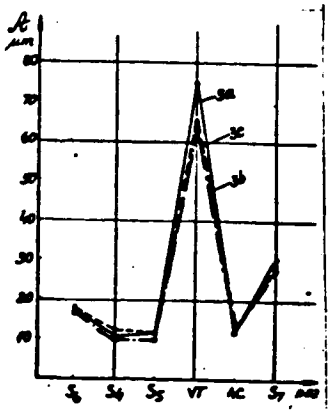

Fig. 2. Vibration amplitude curve for locomotive assemblies

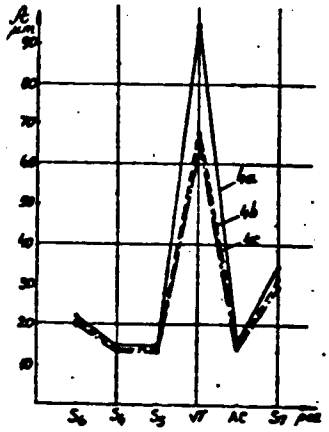

Fig. 3. Vibration amplitude curve for locomotive assemblies

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The notations used on the curves have the following meanings: /78

-- curve 1: locomotive with assemblies in operation;

-- curve 2: individual locomotive in motion with assemblies in operation;

-- curve 3: locomotive hauling passenger cars;

-- curve 4: locomotive hauling freight cars.

The letters that accompany the above numerals have the following meanings:

a) accelerated conditions with $I_{motor} > 0$;

b) idling conditions with $I_{motor} = 0$;

c) braking conditions with $I_{motor} < 0$.

S_4, S_5, S_6 and S_7 denote the unit assemblies, VF denotes the T_{12} ventilator and AC denotes the 36.1 compressor assembly.

Analysis of the measured values discloses that the highest vibration level occurs in the case of the locomotive hauling freight cars under accelerated conditions with $I_{motor} > 0$.

The maximum vibration amplitude A was obtained for the T_{12} ventilator assembly and attains a value of 95 μm.

Thus, in order to reduce the high vibration level that occurs in the assemblies of electric locomotive machine rooms during their operation, it is necessary to reduce the high values obtained in the T_{12} ventilator assembly. We acted in this direction by taking design as well as constructional measures with a view to reducing the vibration level in this assembly, with which we shall not deal in the present study.

In the course of conducting these measurements we felt the need for some standards to regulate the vibration level occurring in trains.

Summary

Because of the high vibration level occurring during the running of electric locomotives, mechanical defects occur in the control mechanisms of auxiliary services. This fact has given rise to the necessity of determining the values of these vibrations in electric locomotives during their running. Measurements were conducted on the Craiova-Drobeta-Turnu Severin line. /80

On the basis of our results we took measures to reduce the vibration level in the electric locomotives manufactured by the Craiova "Electroputere" Factory.

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N74-28372

CONTRIBUTIONS TO THE PROBLEM OF PIEZOELECTRIC ACCELEROMETER CALIBRATION

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Accurate calibration of vibration transducers entails considerable difficulties due to the fact that in producing sinusoidal motions with rigorously known parameters, especially at higher frequencies, as well as in ensuring that the different points of the same body should execute rigorously identical motions, we are confronted with harder problems than might appear at first blush. /81

Known calibration methods can be divided into two great classes:

Absolute methods, which consist in imparting to the transducer a known motion, or in measuring the parameters of motion with a different method from that of the transducer to be calibrated.

Relative methods, which are based on comparison of the signal generated by the transducer to be calibrated with that from a standard transducer under identical conditions of motion.

To the first class belong: the method of releasing a ball at the acceleration of gravity, the microscope method (measurement of the amplitude of motion by means of a measuring microscope) and interferometric methods. The first of these methods is particularly simple but ensures no greater accuracy than 5-10%; the other methods are more precise but require complicated and costly equipment. Special mention should be made of laser interferometry, which ensures an accuracy of 0.5% in the 80-1,000 Hz frequency range.

The precision of comparative methods, which also occur in a number of versions (for example, the insertion method), is obviously limited by the accuracy of calibration of the standard and is, therefore, inferior to that of absolute methods. /82

A special position is occupied by the reciprocity method, which is based on a theorem that generally holds for linear systems and according to which the excitation applied at one point of the system and the response obtained at another point are interchangeable. The reciprocity method can be applied with particular convenience to piezoelectric transcuders, which belong to the category of reversible electromechanical systems. The reciprocity method, as we shall see, does not require complicated mechanical or optical equipment.

Theory of the Reciprocity Method

The piezoelectric transducer can be regarded as an electromechanical quadruple whose sinusoidal operation is described by the equations

$$\begin{aligned} E &= Z_E I + Z_{EM} V \\ F &= Z'_{EM} I + Z_M V \end{aligned} \qquad (1)$$

The equality $Z_{EM} = -Z'_{EM}$ expresses the property of reciprocity.

The impedances Z_E, Z_{EM} and Z_M are determined as follows:

$$Z_E = \frac{E}{I}\Big|_{V=0} = \frac{1}{j\omega C} \| R_{iz} \approx \frac{1}{j\omega C}$$

The insulation resistance of transducers is usually negligible. In the transducers manufactured by the Institute of Electronic Research it is greater than 1,000 Mohm.

$$Z_{EM} = \frac{E}{V}\Big|_{I=0} = j\omega \frac{E}{A}\Big|_{I=0} = j\omega S$$

where S = E/A is the voltage sensitivity of the piezoelectric accelerometer.

$$Z_M = \frac{F}{V}\Big|_{I=0} = j\omega \frac{F}{A}\Big|_{I=0} = j\omega m$$

The property of reversibility of piezoelectric accelerometers arises from Eq. (1):

1. the piezoelectric vibration transducer subjected to a sinusoidal motion with the velocity V and pulsation ω delivers at its terminals a voltage of the same frequency;

2. the transducer excited with a voltage E goes into vibration. /83

For two transducers rigidly fastened to each other, we can write the equations:

$$\begin{aligned} E_a &= \frac{1}{j\omega C_a} I_a + j\omega S_a V \\ -V + F &= j\omega S_a I_a + j\omega m_a V \\ E_m &= \frac{1}{j\omega C_m} I_m + j\omega S_m V \\ -V - F &= j\omega S_m I_m + j\omega m_m V \\ E_m &= -R I_m \end{aligned} \qquad (2)$$

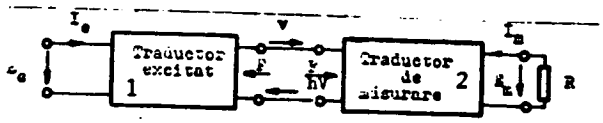

Fig. 1. 1. Excited transducer; 2. Transducer to be measured.

The subscripts e refer to the excited transducer and the subscripts m refer to the transducer to be measured. F denotes the force of interaction between the two transducers.

Taken into consideration in the equations is the viscous damping force hv, as well as the loading effect of the measuring transducer.

From the system of equations for the ratio E_e/E_m we get the relation:

$$\frac{E_e}{E_m} = \frac{S_e}{S_m} + \frac{2k}{\omega^2 S_e S_m C_e C_m R} + \frac{M}{\omega^2 S_e S_m C_e}$$
$$- j\left(\frac{S_m}{\omega S_e C_e R} + \frac{S_e}{\omega C_m S_m R} + \frac{M}{\omega^2 C_e C_m S_e S_m R} - \frac{2k}{\omega^2 C_e S_e S_m}\right) \qquad (3)$$

in which $m_m + m_e = M$.

So by measuring the voltage ratio E_e/E_m we get a relation that links the product of the transducers' sensitivities to the system's other parameters.

For the usual values of the terms that enter into the above equation, the first term has a value on the order of unity, the second, on the order of tens and the third, on the order of 10^6. /84
As a consequence, the first two terms can be neglected. Likewise, owing to the use of a lock-in voltmeter, the terms in parentheses, which represent the E_m component dephased by 90^o as compared with the voltage E_e, cannot be detected. The measured voltage ratio is, therefore:

$$\frac{E_e}{E_m} = \frac{M}{\omega^2 C_e S_e S_m} \qquad (4)$$

From this equation we obtain the product $S_m S_e$ of the sensitivities of the two transducers.

If we determine the products of the three transducers with sensitivities S_1, S_2 and S_3 in pairs, we get the system of equations:

$$S_1 S_2 = A \; ; \quad S_2 S_3 = B \; ; \quad S_3 S_1 = C \qquad (5)$$

whose solution yields the three values S_1, S_2, S_3.

Description of Calibration Setup

The block diagram of the setup is shown in Fig. 2.

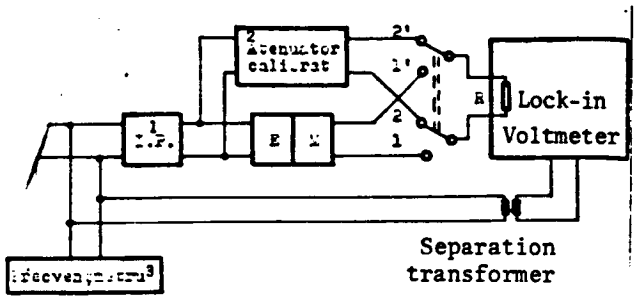

Fig. 2. 1. Phase inverter; 2. Calibrated attenuator; 3. Frequency meter.

The ratio E_e/E_m is measured with the aid of the calibrated attenuator with an accuracy better than 0.3%. Measurements are conducted in two stages:

1. the indication given by the lock-in voltmeter is noted when it measures the voltage E_m.

2. the commutator K is set to position 2-2' and the calibrated attenuator is adjusted until the lock-in voltmeter indicates the same value as at point 1 and the value of ratio E_e/E_m is read on the attenuator. /85

Because of the design of piezoelectric accelerometers, E_m is in antiphase with respect to E_e. The calibrated attenuator does not reverse voltage phase. In order to avoid any error that might be due to possible asymmetry of the lock-in voltmeter, the polarity inverter IP was introduced.

The generator was set to the operating frequency by means of a frequency meter.

As excitation signal level we chose 60 V_{ef}.

The measuring transducer's voltage level was approximately 15 μV. The lock-in voltmeter eliminates fluctuation and network noise and permits accurate measurement even at these voltage levels.

We carried out a careful analysis in order to eliminate mass loops. The synchronization voltage was prevented from penetrating into the measuring circuit by means of a separation transformer.

In building the setup, we used standardized electronic apparatus with the exception of the precision attenuator built by the present authors and a lock-in voltmeter, which was built on the basis of a 2107 Bruel & Kjaer low-frequency analyzer, a numeric Solartron LM14403 voltmeter and a specially designed synchronous detection unit.

In order to obtain correct results, it is necessary that the three measurements that furnish the data for system of equations (5) be conducted at the same temperature.

The capacitance C_e was found to be very dependent on the excitation voltage. For this reason, its measurement at a point was replaced by measurement of the current I_e at the operating voltage itself.

In order to eliminate the effect of vibrations from the environment, the two transducers, which were tightly fastened together, were suspended from a wire extended by a spring. The frequency of /86
the assembly's proper oscillations was approximately 1 Hz in the case of swinging oscillations as well as longitudinal oscillations.

With the aid of the above described setup, we conducted calibration measurements on a number of piezoelectric accelerometers of foreign and domestic manufacture. The results show satisfactory agreement with the sensitivity values provided by the manufacturer or determined by other methods.

Conclusions

Using a lock-in voltmeter for the calibration of piezoelectric transducers by the reciprocity method enhances the method's precision by permitting correct measurement of the very low voltage levels that come into play in this method. The lock-in voltmeter eliminates the error introduced by viscous damping and electrical loading from the input resistance of the measuring setup.

In order to ensure good accuracy, a number of additional precautions must be taken in the course of calibration.

Summary

After discussing the principal calibration methods for piezo- /87
electric accelerometers, the authors describe an experimental setup they designed for accelerometer calibration by the reciprocity method.

The paper shows how the use of a lock-in voltmeter eliminates errors due to viscous damping and electrical loading.

The accuracy which could be obtained and the cares which must be taken are discussed.

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N74-28374

VIBRATION AND NOISE MEASURING INSTRUMENTS BUILT IN THE RSR

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1. Vibration and Noise Measurement

/89

Together with the rapid advances of science and technology, the measurement of mechanical oscillations and of impacts, as well as that of noises, is acquiring ever more importance.

It is known that the measuring methods of acoustics are similar to those used for vibration measurement inasmuch as vibration sources are usually noise sources too. For this reason, it is useful that the vibration measuring instrument -- the vibrometer -- should be useable for noise measurement too (sonometer).

2. Demands Placed on Measuring Instruments. Manufactured Instruments

The parameters of vibratory motion that are of interest in most cases are: acceleration, velocity, displacement and, in some cases, frequency.

The absolutely indispensable device for electrical measurement methods is the transducer.

In vibration measurements transducers are employed that give an output signal proportional to the velocity of the vibration in the case of electrodynamic and electromagnetic transducers, and to the acceleration of motion in the case of piezoelectric transducers.

With the electrodynamic transducer, measurements are correct to a frequency of approximately 1 KHz and an amplitude of ±1 mm (2 mm peak-peak). The mass that loads the object of study can be quite small. /90

With the electromagnetic transducer, measurements are correct to a frequency of approximately 2 KHz. The sensitivity drops when the distance to the object increases. These transducers are most appropriate for the case when the vibrating masses are small.

The TED-1 vibration transducer that we manufacture is of the electrodynamic type with a crystal and has the following principal technological characteristics:

-- sensitivity: 250 $mV/cm \cdot s^{-1}$;

-- frequency range: 10 Hz to 1,000 Hz;

-- maximum admissable run of crystal: ±1.5 mm opposite the median position;

-- mass of the moveable system: 15 g;

-- mass of the transducer: 900 g;

-- overall size: Ø 50 x 140 mm.

The piezoelectric transducer is the most used in vibration measurements because of its robust construction and high technological characteristics. On the other hand, in this case the measuring instrument is more complicated.

The transducer is mounted directly on the object to be measured and, therefore, the transducer's entire mass loads the object.

The N2104 piezoelectric vibration transducer that we manufacture has the following principal technological characteristics (typical values):

-- voltage sensitivity: between 2 and 5 $mV/m \cdot s^{-2}$;

-- load sensitivity: between 2 and 5 $pC/m \cdot s^{-2}$;

-- insulation resistance: greater than 1,000 Mohms;

-- mass: approximately 30 g;

-- overall size: Ø max. 18 x 30 mm.

These parameters have exact values that are specified in the individual calibration card that accompanies each model.

Calibration accuracy is ±6% and the frequency at which calibration is effected is 275 Hz. /91

An electrodynamic transducer that we manufacture is the portable VP-3 vibrometer. This transistorized apparatus with autonomous supply, together with the electrodynamic TED-1 transducer, forms an independent measuring system that is recommended for operative measurement of vibrations in machines, assemblies and construction elements. The kit that is delivered contains the transducer, the electronic measuring instrument, the rectifier for charging the accumulator and the different accessories for fastening the transducer.

The principal technological characteristics of the electronic device are:

-- frequency range: 10 Hz to 1,000 Hz;

-- vibration amplitude measurement range (peak values): 1 to 1,000 μm;

-- vibration velocity measurement range: 0.1 mm/s to 100 mm/s;

-- outlet for the oscilloscope;

-- supply: 9.6 V Cd-Ni accumulator rechargeable from the network;

-- overall size: 235 x 163 x 104 mm;

-- mass: approximately 2.8 kg.

For the piezoelectric transducers the electronic N2103 vibrometer was conceived, which uses in the input circuits transistors with a field effect. The device has a modular construction with interchangeable drawers so that it can be used as a vibrometer or as a sonometer.

The preamplifier for vibration measurement is used in two operating modes: as a voltage preamplifier and as a load preamplifier. The operating mode is chosen by means of a button on the front panel of the preamplifier drawer.

In order to obtain a good response at low frequencies, it is necessary for the product of the resistance R and the capacitance C of the circuit of the preamplifier input to be as great as possible. The voltage preamplifier must have at the input as great a resistance R as possible (in the N2103, R is approximately 1 Gohm). With the load preamplifier an output voltage is obtained that is independent of the input circuit's capacitance, so that a transducer-apparatus connecting cable of any length can be used without leading to an undesirable change in the measuring system's sensitivity. The connecting cable in the N2103 vibrometer with a length of 100 m reduces sensitivity by at most 1% in the event of maximum amplification. /92

The signal from the transducer that is proportional to the vibration acceleration can be measured directly or integrated electrically once or twice in order to measure the velocity or displacement of the vibratory motion.

Other principal technical characteristics of the precision N2103 electronic vibrometer are as follows:

-- direct indication of effective peak and mean values. The measuring circuit indicates the correct effective value for signals with a crest factor (ratio between peak value and effective value) up to 5;

-- overload indicator for signals with a crest factor greater than 5;

-- measurement range: for acceleration: 0.03 m/s^2 to 10,000 m/s^2; for velocity: 0.03 mm/s to 50 m/s; for displacement: 0.03 µm to 1 m;

-- frequency band: 2 Hz to 40 KHz, with the possibility of band limitation upon choice (20 Hz and 10 KHz);

-- outlet for visualization on the oscilloscope;

-- terminals for the external filter. The external filter can be used concomitently with internal filters;

-- supply: 220 V 50 Hz network;

-- size: 470 x 190 x 390 mm;

-- mass: 15 kg.

The apparatus can work with a great variety of piezoelectric transducers including all existing piezoelectric vibration transducers. /93

A N2103 B preamplifier is provided for sonometric measurements. This instrument operates with the precision N2103 electronic vibrometer, constituting drawer 1, with which it forms a precision sonometer. This instrument was built with recommendation CEI 179 for precision sonometers in mind and permits noise measurement with the aid of a condenser microscope with a polarization voltage of 200 V.

The principal technological data are as follows:

-- frequency characteristic: weight curves A, B, C or linear: 2 Hz to 40 KHz;

-- dynamic range: weight A: 40 dB to 148 dB; weight B: 44 dB to 148 dB; weight C: 48 dB to 148 dB; linear: 57 dB to 148 dB. Here we had in view a microphone with a transfer factor of approximately nV/Nm^{-2} (1.5 mV/µbar) (for the reference acoustic pressure a level of 2 x 10^{-5} N/m^2 was considered in accordance with recommendation ISO R 131);

-- the instrument has a dB scale from -6 dB to +10 dB. The measuring circuit, which ensures correct indications in effective

values for signals with a crest factor up to 5 has two response time characteristics in accordance with CEI 179, called Rapid and Slow.

The apparatus can also be used as a low-frequency microvoltmeter (2 Hz - 40 KHz) in the 100 μV - 100 V range.

3. Measuring Instrument Development

Over the last decade, there has been a noticeable tendency toward using transducers that are sensitive to acceleration instead of velocity. Two of the main reasons for this preference are as follows: 1) accelerometers can be built in much smaller sizes than velocity transducers; 2) the useful measurement range of accelerometers is much greater.

For these reasons, instead of the VP3 portable vibrometer /94
that works with velocity transducers, a new type of portable vibrometer called the N2104 is being worked out, which will work with piezoelectric transducers and will measure acceleration, velocity and displacement in a frequency range up to 5,000 Hz. The indication is in effective values. The measuring circuit is calculated to give a correct indication for signals with a crest factor up to 3. The incorporated load preamplifier permits the length of the transducer-apparatus connecting cable to be up to 100 m long without the indication changing by more than 1%.

The apparatus is supplied either by 3 Cd-Ni accumulators or 3 1.5 V R 120 dry cells. The accumulators can be recharged from the network.

In current technology, transitory phenomena and mechanical shocks are often encountered.

Measurement of these shocks requires special measuring and analysis equipment.

In certain cases, the single value that is of interest is the maximum acceleration that takes place during the shock. To this end, we intend to work out special measuring equipment in which provision will be made for using accelerometers and the N2103 vibrometer that have already been built.

For vibration analysis, the precision N2103 vibrometer can be used in combination with a system of filters. The filters will include the normalized A, B, C weight curves used in sonometric measurements, as well as a new weight curve D that is necessary for noise measurement in aviation.

Summary

The authors discuss in brief the demands placed upon vibration and noise measuring instruments. /95

The instruments that are now being manufactured in the RSR are described, as well as those that are being made ready for manufacture, namely: the VP-3 portable vibrometer, the N2103 precision electronic vibrometer, the N2103 B sonometric preamplifier, as well as vibration transducers of the electrodynamic (TED-1) and piezoelectric (N2140) types.

The authors analyze the manner in which this variety of products solves specific problems connected with vibration and noise measurement.

N74-28374

PHOTOELECTRONIC VIBROMETER WITH POLARIZED LIGHT

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The torsion vibrations of shafts, toothed wheels, etc., can be measured by seismic, electromagnetic, optical and other methods [2, 4, 7]. Most of these devices entail some drawbacks, such as for example: the use of sliding contacts, unwieldy construction, necessity of dynamic equillibrations. /97

In the present study, the author will describe an optical vibrometer in which tortion vibrations are transformed into optical signals with the aid of polarized light [1, 4].

The principle of this type of transducer is as follows:

On a transparent celluloid diaphragm a thin film of microscopic crystals from a dichroic substance are deposited and oriented in a certain direction. Such a diaphragm, which is called a polaroid, permits light to pass only for the component of the electric vector $\vec{E}$ whose oscillations are parallel to the optical axis of the polaroid. The incident natural light behind the diaphragm becomes polarized and the electric vectors $\vec{E}$ only oscillate along a single azimuth. This polarized light, as it passes through a second polaroid, will be more or less weakened, depending on the angle α formed between the optical axes of the polaroids.

The principle and block diagram for torsional vibration measurement are shown in Fig. 1.

Parts 1, 2 and 3 rotate together with the shaft whose torsion vibrations are being measured. Polaroid 5, which forms one piece with the inert mass m, has angular deviations with respect to polaroid 3, owing to the shaft's vibrations. The variation of angle α, between the optical axes of the polaroids, modulates the lighting E_2 in the photosensitive transducer 8. /98

How the device is built is shown in Fig. 2.

The light source is a lamp with P = 15 W and U = 12 V.

The lamp is supplied by carbon brushes that press down on the bronze ring 6 drawn across the insulating bushing 7. The thermal inertia of the filament ensures a constant luminous flux even if random fluctuations should occur in the electrical contact resistor. /99

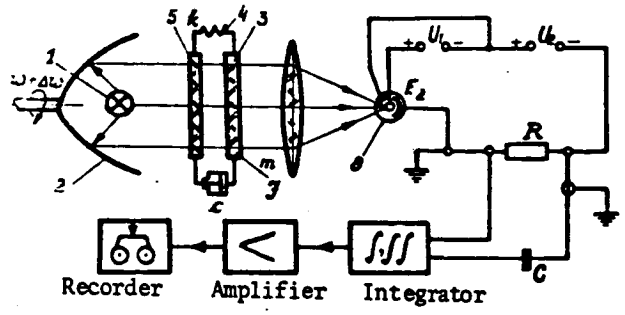

Fig. 1. Diagram of torsion vibration measurement with the aid of polaroids.

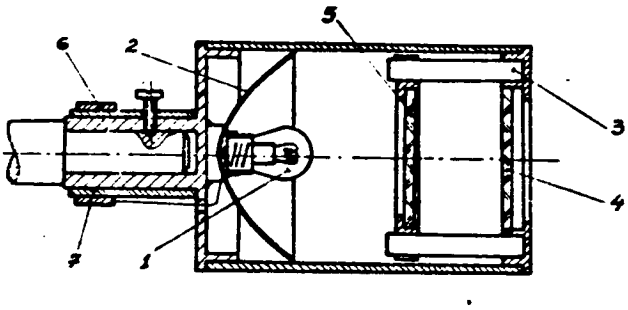

Fig. 2. Diagram of the transducer with polarized light.

Direct light and light reflected by the parabolic mirror 2 pass through polaroid 5 and then through polaroid 3. The mounting of polaroid 5 is held and carried by the elastic coupling of lamellar springs 4.

In order to determine the nature of the device, which can be operated as an accelerometer, a frequency meter or a vibrometer, we will analyze its dynamic operating conditions.

Let us consider a reference vector radius of a uniformly rotating shaft with operating angular velocity ω_1. Because of the noisy running of the shaft, angular accelerations $\pm\varepsilon$ set in. The vector radius of polaroid 5 (or 3) will deviate by the angle ϕ (or ψ), with respect to the reference vector radius (determined by angle $\phi_1 = \omega_1 t$), and the angle of relative rotation between the polaroids is θ.

In order to set up the differential equation of motion for the rotation, we bear in mind that the moment of the forces of inertia is $J\ddot{\psi}$; the viscous damping moment is the result of the relative angular velocity $\dot{\theta}$ and the antagonistic moment due to elastic deformation of the lamellar springs is given by the relative rotation θ between the polaroids.

We get:

$$J\ddot{\psi} + h\dot{\theta} + k\theta = 0 \qquad (1)$$

where: J is the sum of the moments of inertia opposite the axis of the parts that form one piece with the shaft ($kg \cdot m^2$); h_r is the damping coefficient (Nm·s/rad); and k_r is the moment due to deformation of the springs 4 (Nm/rad).

For the radius R the tangential displacements x, y and z correspond to the angles ψ, ϕ and θ. This gives rise to the mechanical model of Fig. 3, in which the motion of real rotation in the apparatus is transformed into the motion of the linear vibrations of the reduced mass $m_{red} = J/R^2$.

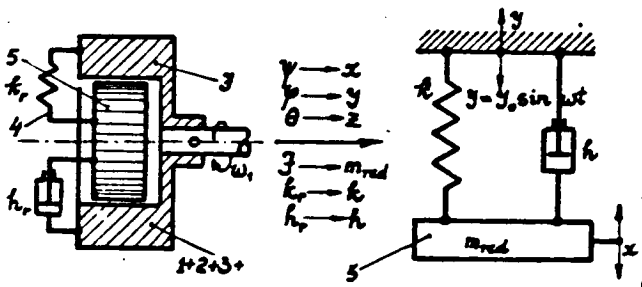

Fig. 3. Simplified model of vibration transducer.

Analyzing the dynamic conditions of this model of seismic devices, the author of reference [2] arrives at the following conclusions: /10

Depending on the value of the ratio between the proper pulsation p of the oscillations of the seismic suspension and the pulsation ω of the vibrations of the shaft, the device can operate under the following conditions [2]:

-- accelerometer: $\omega/p < 0.3$

-- vibrometer: $\omega/p > 3$

-- frequency meter: $\omega/p \cong 1$

The photoelectric transducer's sensitivity depends on the initial angle α_0 between the axes of the polaroids, on the lighting E_2 in the photosensitive element and on the sensitivity of the electronic measuring chain.

The maximum sensitivity for relative rotation $d\alpha$ of the polaroids is obtained for $\alpha_0 = 45^o$. The sensitivity of the system can be increased by using a most intense light source without, however, endangering the polaroid sheet, which is sensitive to heating.

As photosensitive element the following can be used: photodiod, photoresistor, photoelectric cell. At high frequencies it is recommended that a photoelectronic multiplier be used (for example, a FEU-2), which has practically zero inertia and a relatively high sensitivity, S_f = (0.4 ... 0.8) mA/Lx. /10

By judiciously choosing the characteristics of the elements, angular deviations on the order of tenths of centesimal degrees can be brought to light [5].

Summary

The author describes a seismic torsiometer which is based on the reception by a photosensitive transducer of a light flux modulated by a relative rotation of the optical axes of two polaroids. The torsional vibrations of the polaroid fixed to the shaft are transmitted to the other polaroid (which at the same time is the seismic mass of the apparatus) by means of elastic lamellas. /102

The device can work as accelerometer, vibrometer or frequency meter, depending on the value of the ratio between the proper oscillation frequency of the seismic system and the measured vibration frequency.

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N74-28375

VIBRATION CONTROL IN FORGE HAMMERS

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Among measures for ensuring optimum working conditions, control of the noise and vibrations produced by installations, implements and aggregates occupies an important place. /103

Over the last few decades, society has begun to take into consideration in a scientific manner study of the harmful effects of prolonged industrial noise and vibration exposure that is proportional to the number of workers.

In the foundation of an operating forge hammer, vibrations arise as the result of the shocks of the ram against the workpiece or anvil block, vibrations that are transmitted to the working platform, the implements near the hammer and neighboring workshops and offices and that are harmful for the workers operating the hammer as well as for all other persons in the neighborhood.

The vibrations in the foundation of the forge hammer are due to the oscillating system consisting of:

-- ram;

-- anvil block (anvil);

-- foundation block.

Vibrations from this system are transmitted to the working platform and the environment when:

-- the foundation of the hammer is common with the foundation of the room;

-- the couplings between working platform and anvil block are rigid;

-- the construction of the working platform is inadequate or shows a high degree of wear;

-- the foundation block is rigidly placed on the ground;

-- the nature of the ground on which the foundation of the hammer is placed favors the transmission of vibrations. /104

The present paper describes the causes of vibrations as well as a number of solutions for reducing the transmissibility of vibrations to the working platform.

Our analysis shows that even if the hammers are placed on foundations that are separate from the foundation of the room and are equipped with elastic antivibration layers, vibrations are transmitted to the working platform and, therefore, to the worker who operates the hammer, with amplitude values that exceed by far the values permitted by MRPM (Fig. 1). Measurements were conducted on the platform of a 5 t hammer (points 1, 2, 3, 4; point 2 is situated at the place where the worker controls the hammer's operation).

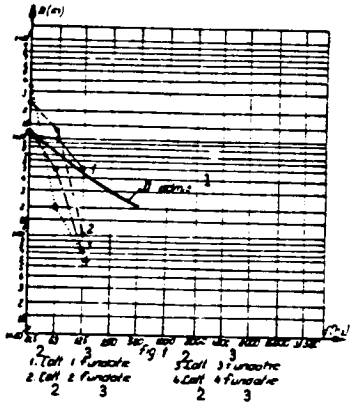


Fig. 1. 1. Permissible; 2. Angle; 3. Foundation.

The noxious effect of these high-vibration levels impairs the health of the workers near the hammer. The amplitudes of the transmitted vibrations exceed the permissible standard curve by values ranging from $5 \cdot 10^{-5}$ cm to $3 \cdot 10^{-4}$ cm in the low-frequency range (31.5 ... 125 Hz).

The motion of the hammer elements was studied with the aid of the equivalent circuit shown in Fig. 2.

The statistical calculation comprises calculation of the weight of the foundation block and of the ram's free-falling energy.

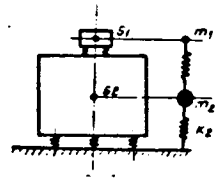


Fig. 2.

For damping the shock effect of the ram's impact, the weight of the foundation block must be as great as possible and the foundation block must be situated on an elastic antivibration layer. /105

The weight of the foundation block G_2 and the value of the ram's free-falling energy E were calculated.

Then a dynamic calculation was conducted, by means of which the following were obtained: the value of the ram's falling velocity during the shock, the velocity of the anvil block during the shock and the velocity of the foundation block.

The design data yielded the anvil block's bearing surface. Between this surface and the foundation block provision was made for a 10 cm hammer felt layer, for which we calculated the elastic

constant K_1(daN/cm) and the pressure that is exerted on it (daN/cm^2). Likewise calculated for the above-mentioned layer was the static flexure.

The designed foundation was elastically borne by the ground. In this case, we calculated the elastic constant for axial displacements since the elastic layer consisted of boxes of imported steel springs. The number of boxes with springs necessary under the foundation block was also determined from design conditions.

For these springs we calculated:

-- the static load P_S on a spring;

-- the tangential unit effort;

-- the elastic constant K_z for axial displacements;

-- the flexure of the spring that will take place under the influence of the static load P_s.

In the nomogram, the proper frequency of the springs that corresponds to this flexure is 3 Hz, and the number of shocks per minute is 180. If we take into account the hammer's maximum beat frequency (80 shocks/min), it turns out that the proper frequency of the springs is not a whole multiple of this value and, therefore, no resonance can take place. Likewise calculated was the number of shocks that correspond to the proper frequency (173 shocks/min), which means that the value of the number of calculated shocks is very close to that found in the nomogram (180 shocks/min).

Then we calculated the substitute static forces:

-- the substitute static force on the anvil block P_1; /106

-- the maximum pressure exerted on the concrete block σ_1;

-- the maximum static force on a spring $P_{spring\ max}$;

-- the dynamic force exerted on the springs P_d;

-- the real substitute static force exerted on the concrete block P_2;

-- the real overload on a spring under a dynamic load without taking into consideration the fatigue coefficient P'_d;

-- the pulsation p and the velocity v of the foundation block in the case of free harmonic oscillations.

In order to reduce the amplitude of motion, whose value was obtained by calculation, we used "VISCAL" dampers.

The force P_1 transmitted by the bed plate of the anvil block acts on the foundation block, and equilibrium is restored by the forces of inertia of the concrete block, forces that are uniformly distributed per section (Fig. 3). We also calculated the maximum bending moment in the longitudinal and transversal directions, as well as the shearing force delimited by the perimeter of the anvil block.

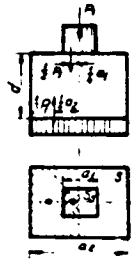


Fig. 3.

We also calculated the resistance of the elastic layer under one of the hammer platform's beams on the assumption that on this beam at a given moment the following many act:

-- the dead weight of the beam;

-- the weight of the upper and lower platforms;

-- the weight of the workpiece;

-- the weight of the workers.

Consequently, we calculated:

-- the total weight (the proper weight of the beam plus the weight of the upper and lower platforms);

-- the maximum load on the beam end;

-- the bearing section of the beam;

-- the unit effort per beam section;

-- the elastic constant and static flexure of the felt layer. /107

Calculations showed that the unit effort and the static flexure are small, which means that the elastic layer we chose was good.

As a result of the entire calculation, for the attenuation of existing vibrations the following solutions were proposed:

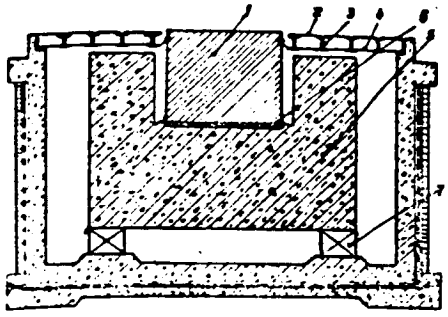
-- reduction of the spaces between the supporting beams of the platforms by doubling the number of existing beams, which eliminates the possibility that static flexures might occur;

-- covering of the spaces that will result after doubling of the beams with sheet metal welded to the lower wings of the beams,

creating a lower platform parallel to the upper platform and located opposite it at a distance equal to the height of the beams;

-- execution of grooves along the entire perimeter of the anvil block so that the hammer scale that flows through the joints between anvil block and platform will collect between the two platforms, from where it can be evacuated periodically. To the same end, provision was also made for inclining the metal sheets of the lower platforms by approximately 45° in the vicinity of the grooved section of the anvils.

These solutions are shown in Fig. 4.



1. Anvil block
2. Upper platform
3. Supporting beam
4. Lower platform
5. Concrete block
6. Felt layer
7. Boxes with springs

Fig. 4.

Summary

The paper discusses special measures to be taken when calculating, designing and executing a forge hammer foundation, so that the vibrations that occur during its working will not be transmitted to neighboring machinery, workrooms and offices because these vibrations are harmful to the workers near the forge hammer.

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N74-38376

VIBRATION AND RECOIL CONTROL OF PNEUMATIC HAMMERS

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1. Introduction

The harmful effect of vibrations on the human body have not yet been completely elucidated because of the complexity of such a study and the fact that research cannot be conducted on parts of the body taken separately in order to deduce the effect of vibrations on them. Results obtained up to the present time clearly show that vibrations have a number of harmful effects of an objective as well as subjective nature. Vibrations can produce physiological as well as physical effects, the most important ones being mechanical and thermal effects. /111

In general, workers who are subjected to the direct action of vibrations exhibit three stages:

a) The first stage may not contain spasmic symptoms although vasomotor disorders already exist. In this stage we find a slight impairment of the fingers' sensitivity, indisposition, uneasy sleep, etc. Working with the hammer, the worker experiences no unpleasant sensations, but on the contrary, feels good.

b) In the second stage, the vasomotor disorders are accompanied by arthrophic disorders, the shape and color of the fingers change, the patient is irritable, complains that he tires easily, feels weak. After special treatment and rest, the worker recovers and can go back to work.

c) In the third stage of the disease, organic changes of the tissues can be observed, as well as muscular atrophy, contractions and temporary changes of the bones. Under the influence of low temperatures or when pressing the fingers into the palm of the hand, spasms accompanied by a sharp and prolonged pain occur. In this stage, the worker's health improves after prolonged treatment and rest, but he cannot go back to work and must avoid any further vibration effects. /112

2. Vibrations and Recoil of Pneumatic Hammers

During the operation of pneumatic hammers, body and tool are displaced, depending on the workpiece, and the value of this displacement depends on the applied pressure and the parameters of the pneumatic hammer. The greater the air pressure, the greater

the hammer's recoil for one and the same pressure and the greater the pressure that must be applied on the handle for the hammer's operation to become stabilized and for basic characteristics, including shock frequency, to be obtained.

A number of studies of machines with percussive action have demonstrated that the properties of the workpiece have an influence on the internal processes and lead to a variation in the parameters. It is known that the rebound (or reflection) velocity is one of the principal characteristics of the percussive piston's return stroke. The element that brings to light the influence of the workpiece on the working of the hammer is the piston's recoil coefficient after percussion. Experiments conducted on different workpieces have shown that the greater the modulus of elasticity, the greater the piston's reflected (or recoil) velocity.

The influence of the pressure on the energy characteristics of pneumatic hammers is different for different design characteristics.

In order to be able to evaluate the value of the force of advance that ensures the hammer's normal working, it is necessary to bear in mind the elements shown schematically in Fig. 1. The body of the working tool is introduced into part B of the cylinder. In order to ensure transmission of the mechanical energy to the tool 2, it is necessary that pressure Q be applied to the hammer. If the pressure is insufficient, the hammer moves away from the pick collar and the percussions of the piston 1 become applied to the cylinder's shoulder 6. Such a mode of operation is inadmissible, so that in order to ensure normal operation, it is necessary to apply a mimimum pressure Q_{min} to the hammer. Denoting by F_1 the force that arises as a result of the piston's percussion against the pin of the tool; F_2, the force of contact between hammer body and pick collar; F_3, the resultant force that acts on the piston owing to air pressure; F_{fr}, the resultant of the resistant forces that appear at the surface of contact between pick pin and wear bushing, and conducting the appropriate calculations, after a series of transformations we get:

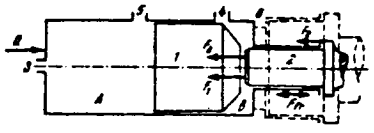

Fig. 1.

/113

$$Q_{min} = 0.4\ P_0\ (0.3\ D^2 - 0.7\ d^2)$$

where P_0 is the air pressure in the network; D is the diameter of the cylinder and d is the diameter of the pick pin.

3. Recoil and Vibration Control of Pneumatic Hammers

Hammer vibrations are characterized by a frequency spectrum. The harmful effect of vibrations on workers is influenced not only by frequency, amplitude and acceleration, but also by other factors such as vibration duration, static muscular tension, position of the body during work, weight of the hammer, ambient temperature, physiological state of the worker, etc.

The principal ways of limiting pneumatic hammer vibrations are: a) changing the shapes of the pressure curves in the two parts of the cylinder; b) introducing elastic damping elements between the hammer's handle and body; c) using antivibration supports and special damping gloves; d) damping of the tool's percussions against the body of the hammer. /114

In the present study, we shall lay more stress on the second method of attenuating vibrations. In solving the problem of insulating the worker from the vibrations of the hammer's head we tried to replace the real man-hammer-environment system by a so-called mechanical model, such as for example, "elastic coupling-mass-stop" system (Fig. 2 a) or the "mass-elastic coupling-mass-stop" system (Fig. 2 b) and the mechanical system with a dynamic oscillation absorbant (Fig. 2 c). In order to improve vibration insulation, we must introduce between handle and body of the hammer of mass m_1 (Fig. 2 b) the additional mass m_2 which is connected to the hammer's body by means of elastic element C. By correctly choosing mass m_2 and the rigidity of the elastic element, we can reduce the handle's oscillation amplitudes below the limits permitted by the standard.

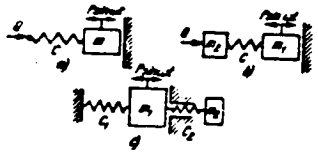

Fig. 2.

In the case of the mechanical model with a dynamic oscillation absorbant (Fig. 2 c), an additional mass m_2 is attached to the body of the hammer of mass m_1 by means of elastic coupling C_2. The rigidity of the coupling C_2 and the magnitude of the mass m_2 are chosen in terms of the resonance condition at which the oscillation amplitude of mass m_2 increases sharply and the vibrations of the body of mass m_1 decrease.

Theoretical studies and experiments have demonstrated that in the case of calculating antivibration devices it is well to pass through the following stages:

a) Determination of the weight of the vibration damper's body and handle.

b) Choice of the adimensional frequency that determines the minimum values of the handle's oscillation amplitude.

c) Determination of the rigidity coefficient of the elastic coupling.

d) Determination of the spring's size and its characteristics. /115

Mentioned in the specialty literature are several types of pneumatic hammers equipped with antivibration devices. Thus, Fig. 3 shows a hammer of the Leontiev type where 1 is the handle; 2 is the intermediate ring; 3 is the support for the spring; 4 is the clamping sleeve; 5 is the spring; 6 is the guiding cylinder for the spring; 7 is an intermediate element; 8 is the distribution box; 9 is the distribution device; 10 is the cylinder; 11 is the damper; 12 is the joining wedge; and 13 is the piston.

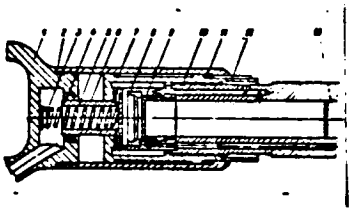

Fig. 3.

Fig. 4 shows the MO-1 model, where the handle is insulated from vibrations by means of the spring 4. The handle 1 slides along the distribution body 2, being guided by the ring 3. The elastic coupling between the handle's sleeve 5 and the cylinder 6 is effected by means of the spring 4.

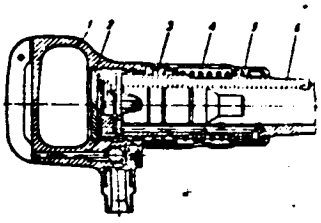

Fig. 4.

perfected are the MRV-12 (Fig. 5) and MRV-16 (Fig. 6) models [1]. The characteristics of these models are the presence of the device 7-8-9 for damping the vibrations transmitted from the pick, the use of the conic spring 10, ... [translator's note: next few words are illegible] of the hammer by means of the elastic hose 11 and the presence of the false cylinder 2. Researchers in our country, too, have been preoccupied with the creation of pneumatic hammers with a reduced vibration level. /116

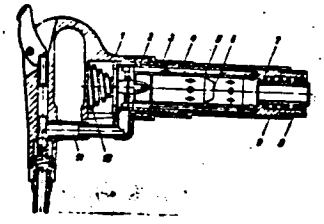

Fig. 5.

Thus, a group of authors has calculated and designed a new type of pick hammer (Fig. 7) that has the following advantages over known types: a) a much simpler vibration damping device consisting of a mobile cover 5; b) the hammer is equipped with self-distribution, the piston distributing the air in both parts of the cylinder; c) lower air consumption as a result of air expansion phases; d) smaller number of component parts.

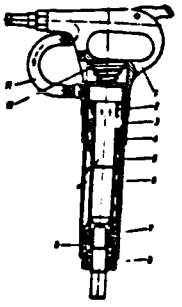

Fig. 6.

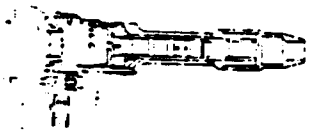

Fig. 7.

Summary

The authors describe the principal vibration sources of the pneumatic hammers used in the mining industry (pick hammers), in boiler shops (riveting hammers), etc., bringing to light the fact that the principal vibration source is the variation in air pressure inside the cylinder. Then, the authors discuss the present state of the art of vibration control of pneumatic hammers as it is practiced abroad and the solutions adopted for this purpose. /11

In the last part of the study, a new type of pneumatic hammer with a low noise and vibration level is presented.

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OBSERVATIONS ON SOME ACOUSTIC METHODS USED IN STUDYING THE ELASTIC PROPERTIES OF METALS

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1. Introduction

The propagation of elastic vibrations in a solid medium of finite dimensions are influenced by the limiting surfaces of the medium in question and by its anisotropy. The presence of limiting surfaces in an isotropic solid medium requires that certain limiting conditions be met. In general, the existence of the lateral surface means that the vibration will not be purely longitudinal. It can be regarded as a quasilongitudinal vibration, hence, a vibration with a weak transversal component. /119

Inasmuch as solutions of the equation of elastic vibrations that satisfy all the limiting conditions come up against considerable mathematical difficulties, probable solutions will be studied for a long cylindrical beam (length $L \gg 2a$, where $2a$ is the diameter).

The quasilongitudinal manner of vibration along the axis of a long cylindrical beam of length L and diameter 2a can be studied with the aid of the mathematical formalism of Pochhammer [1] and Ezz-El-Arab [2]. Because the excitation of a solid medium in the low-frequency range is simple and easy to effect in practice by using acoustic resonance methods, our experimental measurements were conducted at the fundamental vibration frequencies of the audio range.

Comparison of our data with those of the specialty literature lead to some interesting and useful observations.

2. Theoretical Considerations

Study of the symmetric quasilongitudinal vibrations of a long, small-diameter, cylindrical beam can be simplified if we consider a quantity w called the elementary displacement. It is associated with the small elastic motions, variable in time, that any material particle of the solid medium in question executes. Because of the cylindrical beam's symmetry of rotation about the oz axis, w depends only on the r and z coordinates of the position of the particle under consideration. In the case of zero cubic expansion of the solid medium, the equation of the small, symmetric, elastic displacements can be written in the form: /120

$$\frac{\partial^2 w}{\partial r^2} + \frac{1}{r}\frac{\partial w}{\partial r} + \frac{\partial^2 w}{\partial z^2} + \frac{4\pi^2 f^2}{(v')^2} w = 0, \qquad (1)$$

where: f is the frequency of the quasilongitudinal vibrations and v' is their velocity.

Inasmuch as the vibrations we have called quasilongitudinal also contain a weak transversal component (because of the Poisson effect) velocity $v' = v/f(\sigma)$, where v is the velocity of the longitudinal oscillations and σ is Poisson's ratio. Obviously, $v = (E/d)^{1/2}$ (E is the modulus of elasticity or Young's modulus and d is the density of the solid medium) is obtained in the case when the value of the function $f(\sigma)$ is practically equal to unity; $f(\sigma) \gtrsim 1$.

A possible solution of Eq. (1), an equation of the Euler-Bessel type with a zero subscript and a parameter, would be

$$w = A J_0(hr) \sin (h^2 - b^2)^{1/2} z, \qquad (1a)$$

where the letters have the following meaning: J_0 is the Bessel function of the order zero, A and b are real constants and $h = 2\pi f/v'$.

For the limiting conditions we distinguish two important cases: b equal to zero and b different from zero.

In the case of $b = 0$, since the relations $hL = n\pi$ ($n = 1, 2, 3, \ldots$) and $f(\sigma) = 1$, the vibrations become longitudinal. The proper frequency of the cylindrical beam will be given by the known relation:

$$f = nv/2L = (E/d)^{1/2} \cdot n/2L. \qquad (2)$$

As a result, only for $b = 0$, when the transversal component is negligible, can the beam's vibrations be considered longitudinal.

When the diameter 2a begins to increase (L = const.) or the length L decreases (2a = const.), for a certain ratio L/2a the elastic vibrations will be influenced by the transversal component. In this case, b will assume values different from zero. /121
For small values of b (bh smaller than unity) the condition:

$$hL = n\pi (1 + \Delta), \quad \Delta \ll 1$$

is satisfied.

This relation leads to the expression for Δ for $n = 1$ and L variable that expresses the fact that the beam's new resonance

frequency f' will be smaller than the preceding frequency given by Eq. (2), namely:

$$\Delta = \frac{f - f'}{2} = \frac{\Delta f'}{2}, \tag{3}$$

where $f' = v/2L\ f(\sigma)$, $f(\sigma) > 1$.

Obviously, beginning with a certain ratio L/2a, the transversal component will influence, to a greater extent, the cylindrical beam's quasilongitudinal elastic vibrations and an inertia effect of the transversal displacements will, therefore, take place. Owing to this effect, the beam's proper frequency will increase in the proportion [3]:

$$f : (f + \Delta f) = 1 : (1 + \pi^2 a^2 \sigma^2 / 4L^2).$$

Thus, we get the relation:

$$\frac{\Delta f'}{f} = \frac{f' - f}{f} = \frac{\pi^2 a^2 \sigma^2}{4L^2}, \quad (\sigma > 1). \tag{4}$$

Equation (4) permits determination of Poisson's ratio in the most convenient case, namely, when the diameter 2a is kept constant and the beam's length L decreases. The expression for σ is:

$$\sigma = (\Delta f'/f)^{1/2} \cdot 2L/\pi a, \quad (a = \text{const}) \tag{5}$$

Eqs. (2) and (5) make it possible to determine the elastic constants E and σ that characterize the material's elastic properties in the low-frequency range.

3. Experimental Results

The connecting diagram of the apparatus we used in our experimental determinations is as follows:

Oscillator -- exciting transducer (I) -- test piece -- receiving transducer (II) -- voltage amplifier -- oscilloscope followed by an electronic voltmeter and a numeric frequency meter.

The manner of operation is based on a known acoustic resonance method.

The oscillator RC supplies a low-power (1 m W) sinusoidal voltage that can be adjusted for amplitude (0-10 V) and frequency. The electric oscillations are transformed into the elastic vibrations of the test piece by means of the electromagnetic transducer (I). The test piece, which is fixed in the /122

middle, vibrates strongly for the fundamental frequency (proper oscillation frequency).

The electric oscillations induced in the electromagnetic transducer (II) are applied to the voltage amplifier with an amplification factor that can be adjusted within the limits of $0\text{-}10^4$. At the latter's output we connected concomitantly the vertical deflection plates of the cathode ray oscilloscope, the input of the electronic voltmeter and the 0.1 Hz input of the precision numeric frequency meter.

The experimental results obtained for the proper audio frequency band are shown in Table I.

TABLE I

Material	L (m m)	2a (mm)	f (Hz)	Δf (Hz)	v (m/s)	E (Kg.10^{-5})	f' (Hz)	Δf' (Hz)	σ
Brass	500	11,0	3495,5	0,5	3495,5	10,37	-	-	-
Copper	500	12,0	3677,2	1,8	3677,2	12,00	-	-	-
Macled zinc	255	14,7	7870,0	6,0	4013,7	11,60	-	-	-
Steel (OL 38)	500	12,5	5165,7	1,0	5165,7	20,45	-	-	-
Steel (OL 38)	400	12,5	6457,1	-	5165,7	20,48	6455,0	4,1	-
Steel (OL 38)	350	12,5	7379,1	-	5165,7	-	7385,8	4,5	0,429
Steel (OL 38)	300	12,5	8609,5	-	5165,7	-	8615,8	4,1	0,334

To calculate the elastic constants E and σ we used Eqs. (2) and (5), respectively.

4. Experimental Data Recorded in the Literature

The region that lends itself best to different resonance methods is the low-frequency region ($\lambda \gg 2a$), which has been investigated by many researchers [4, 5, 6, 7, 8, etc.].

Recently, Glandus and Boch [5] determined the modulus E and the coefficient σ by a method of resonant beams (test piece with 2a: 10 mm and L: 150 mm) working on odd harmonics (n = 3, 5, 7) of the fundamental frequency. These authors showed that the Poisson effect introduces between the odd harmonics of the fundamental frequency and the experimental values a discrepancy that permits the value of the elastic constants E and σ to be determined by means of equations derived from Pochhammer's solution. /12

Table II shows some of the values obtained by the above-mentioned authors as compared with those from recent tables of physical constants [5] (Smithelle; AJPR; Truel and Elbaum).

TABLE II

Material	E (10^{10} Pa)	σ	Tables of physical constants E	Tables of physical constants σ
Steel (XC 65)	21,16	0,299	20,14 - 21,19	0,276 - 0,296
Duraluminum	7,15	0,344	7,00 - 7,15	0,335 - 0,345
Brass (70-30)	8,67	0,359	10,00 - 10,70	0,350 - 0,374

The resonance method based on determination of the half width f of the resonance curve [6] permits the modulus E and the internal friction coefficient Q^{-1} to be determined. Experimental measurements [6] conducted on test pieces of gray cast iron (2a: 12 mm and L: 350 mm) of different qualities for n = 1, are recorded in part in Table III.

TABLE III

Sample Number	f (Hz)	Δf (Hz)	Logarithmic Decrement	Q^{-1}	E (kgf/mm²)	Remarks
2	5755	45	0,0135	0,4296	11944	-
4	5974	40	0,00911	0,2899	12906	Superior quality

Also significant are the experimental results shown in Table IV, likewise obtained by acoustic resonance methods [7, 8].

5. Discussion

It can be seen from the data of Table I that, in the first place, the investigated metals show very little damping (their internal friction is also little) -- with the exception of macled zinc. In the second place, the data shown in the last part of Table I confirm the present author's hypothesis about the influence of the transversal component of quasilongitudinal vibrations. /124

As a result, Eqs. (3) and (4) are verified experimentally within the limits of the measurement error.

TABLE IV

Material	L (m)	2a (mm)	f (Hz)	v (m/s)	v (m/s)	Reference
Steel (38)	1,5e	5,oo	17o8	5124	51oo (x)	[7]
Copper	1,51	5,oo	1558	5558	55oo (x)	[7]
Copper	o,98	5,oo	184o	5466	55oo (x)	[7]
Aluminum	2,oo	7,oo	116o	464o	5ooo (x)	[7]
Wood (beech)	1,5-	8,oo	15o6	5918	59oo (x)	[7]
Wood	1,5o	8,oo	148o	5848	59oo (x)	[7]
Wood	1,1o	8,oo	18oo	5964	59oo (x)	[7]
Steel	o,9144	6,35	28oo	512o	513o (xx)	[8]
Copper	o,9144	6,35	178o	5255	556o (xx)	[8]
Aluminum	o,9144	6,35	2675	489o	51o4 (xx)	[8]

(x) Table 2.1 of ref. [9]
(xx) Smithsonian Tables [8]

It can be seen from Table III that Q^{-1} decreases with increasing E and the quality improves. Micrographs of test pieces 2 and 4 attest to the fact that the decrease in length and diameter of graphite separations [6] causes the drop in internal friction (the value of E).

Finally, Table IV confirms the hypothesis that only for $L \gg 2a$ can the velocity be considered equal to v.

It should be recalled that $b^2 = \Delta \cdot 4\pi nh/L$.

6. Conclusions

Resonance methods in the low-frequency band facilitate determination of both elastic constants (E,σ) only if we adopt the method recommended and verified in the present study. Further, between the values obtained and those cited no significant differences are introduced. This fact argues in favor of using the audio frequency range for practical purposes inasmuch as experiments can then be undertaken in a simpler manner.

Summary

The authors describe an experimental setup that permits very /125 accurate measurement of the resonance frequencies of long cylindrical beams fixed in the middle and whose size can vary within wide limits. It also permits measurement of the width of the resonance curve (f = 0.7f). It was found that the Poisson effect can be brought to light for relatively long beams by Eq. (3) and for relatively short beams, by Eq. (5).

With the aid of Eq. (5) the authors calculated the Poisson ratio of the examined material.

The values obtained with this method argue in favor of using the low-frequency region for practical purposes of determination of the elastic constants of solids.

C^2

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ACOUSTIC DETERMINATION OF CRACKS IN WELDED JOINTS

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One of the difficult problems that is frequently encountered in welding metals is the appearance of cracks, especially in the region of the basic metal that is close to the welding cord. Recently used for this purpose is apparatus based on acoustic analysis [1, 2, 3, 4, 5]. The method consists in analyzing, on the one hand, the vibrations generated in the metal part that is being welded at the moment a crack appears, and, on the other, the crack's propagation in time. The experiments in the specialty literature were conducted on samples involved in research on cracking in welds, such as the CTS research and that of the U.S. Navy's Research Laboratory for determining the cracking sensitivity of the melted region [1] or on cross-shaped samples in the case of delayed cracking [6]. /127

Our experiments deal with determining the welding technology for a ferritoperlitic gray cast-iron part that we proposed to remedy by welding. In view of the difficulties raised by possible preheating, we arranged for the part to be cold welded. To this end, we used QK 52-98 electrodes of ESAB manufacture with a diameter of 3.25 mm. Electrode composition was 55% Ni and 43% Fe. The length of the welding zone was 95 mm.

Welding conditions were as follows:

-- arc voltage U_s = 22-24 V;

-- welding current I_s = 90-100 A;

-- welding rate V_s = 0.0368 - 0.0638 cm/s.

As can be easily seen in view of the great size of the area affected by defects, the welding rate was extremely slow, which led to our using high linear energies, with values ranging from 25.6 to 44.5 kJ/cm. In view of electrode consumption, a welding line consisted of four welding cords. After completing one line, the welding direction was reversed. /128

The acoustic emission pickups were KD 35a piezoelectric accelerometers with magnetic fixation on the workpiece. As integrating amplifier we used a SM 241 device, and we used a direct-recording oscillograph to record the phenomenon with an 8 LS-1 galvanometer. All the apparatus was manufactured by the RFT (German Democratic Republic).

Fig. 1 shows the workpiece to be welded and the manner of placing the piezoelectric vibration transducers. The hatched area refers to the region that was remedied by welding.

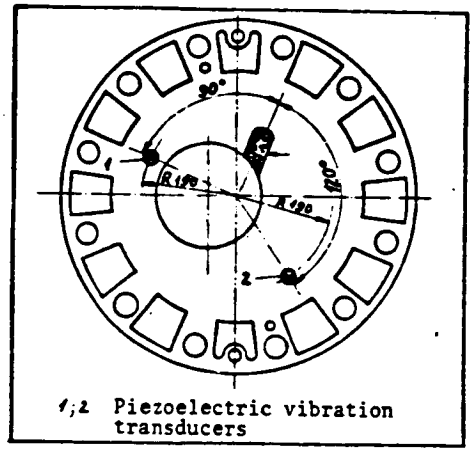

Fig. 1. Workpiece to be welded and position of the vibration transducers.

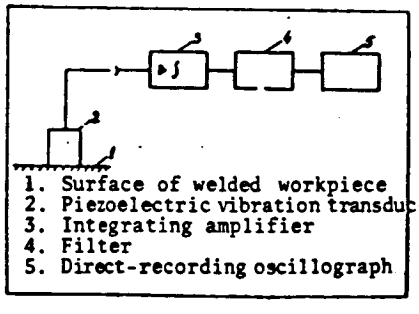

Fig. 2. Connecting diagram for measurements.

The connecting diagram /12 of the measuring setup is shown in Fig. 2, and it holds good for any transducer. In the case of our experiments, we used two transducers whose position on the workpiece can be seen in Fig. 1.

On the recordings obtained during effective welding as well as after welding (apart from the time for cleaning off the deposited slag layer by hammering) we observed the following (Fig. 3):

-- a background noise that remains relatively constant throughout the course of of recording for both transducers;

-- the appearance of peaks at different intervals and amplitudes, which indicates crack formation;

-- transducer 2 is more sensitive than transducer 1 for the frequency and amplitude of defect peaks;

-- defects appear with the first welding deposits;

-- in the last two welding lines the defects are brought to light for frequency and amplitude to a lesser extent than in the preceding lines;

-- after complete welding, during the cooling of the deposited metal we can also observe the appearance of smaller and greater amplitudes, which testifies to /13 the appearance of defects;

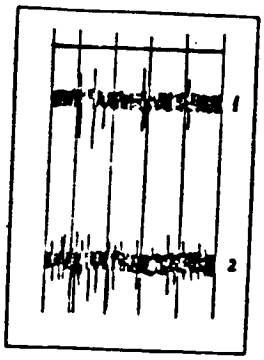

Fig. 3. Copy of recording obtained.

-- the quantitative value of the accelerations determined on the basis of our measurements are shown in Table 1.

From the foregoing the following conclusions can be deduced:

-- the appearance of defects is more pronounced in the first deposits, which is in agreement with normal welding phenomena;

-- plainly to be seen with the naked eye are cracks at the exterior of the welding cord in the region indicated by the arrow in Fig. 1, which in favor of a greater sensitivity in the case of transducer 2, in view of its position;

-- the method is sensitive and convenient /131
for use in particularly extensive weldings or for determination of appropriate welding conditions in the case of working.

TABLE 1: DETERMINED ACCELERATION VALUE

Piezoelectric transducer	Acceleration m/s^2	m/s^2
	Background	For defects
1	2.3	6.5
2	3.0	9.0

Summary

Welding technology must be established on the basis of /133
experiments in order to avoid defects that endanger the durability of the welds. One of the most serious defects that can occur in welded parts is cracking.

The bringing out of cracks, especially in the region of the basic metal that is close to the welding cord, is difficult by means of nondestructive testing methods.

The acoustic analysis method has recently permitted detection of any cracks that might take place and their manner of propagation.

The study deals with the cracks produced in experiments to determine the welding technology for a welded gray cast-iron

workpiece. For this purpose, the authors used piezoelectric transducers to determine vibration acceleration.

The results obtained suggest the possibility of studying these cases by means of the proposed method.

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N74-28379

CONSIDERATIONS ON THE ACOUSTIC ENERGY RADIATED BY TOOTHED GEARS

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Timisoara "Traian Vuia" Polytechnic Institute

Gears with toothed wheels form part of a kinematic chain consisting of the driving motor, coupling, transmission belts, shafts, bearings, housing, etc. Complete study of vibration and noise generation in transmission elements requires dynamic analysis of the entire system [1, 3]. The latter forms a complex of coupled partial oscillating systems. /135

Calculation of the acoustic energy radiated by a running gear entails considerable difficulties [2, 6, 7]. Because of the multitude of factors, insufficiently known parameters and simplistic hypotheses, theoretical results generally provide only rough data for the level of noise generated by the gear.

The effect of the shock vibration phenomena that take place during the gearing of gear teeth is propagated in the form of an air sound and a body sound. From the practical point of view, it is important to know what is the ratio of the acoustic energies transmitted by these two waves in forming the acoustic field around the gear housing.

Fig. 1 shows the mechano-acoustic model proposed in ref. [4].

According to this model, the gear is regarded as an acoustic doublet consisting of the two running toothed wheels. The teeth of any wheel generates cylindrical acoustic waves, but the frontal surfaces of wheels behave like pulsating pistons that radiate acoustic energy into the surrounding space in the form of spherical waves.

The tridimensional differential equation of acoustic wave propagation expressed in Cartesian coordinates has the form: /136

$$\nabla\Phi = \frac{1}{c^2}\frac{\partial^2\Phi}{\partial t^2} \qquad (1)$$

where ∇ is the Laplace operator, c is the velocity of the acoustic wave and Φ is the velocity potential.

After transformation of Eq. (1) into cylindrical or spherical coordinates, we get Bessel-Neumann differential equations. By solving these equations we get the following expressions for the

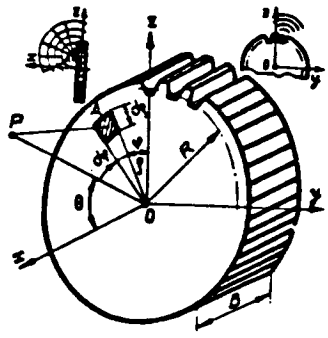

Fig. 1. Mechano-acoustic model of gear with toothed wheels.

acoustic pressures d_c and p_s related by the progressive cylindrical or spherical waves generated by the vibrations of the toothed wheels [4]:

$$p_c = \rho_0 V_0 \sqrt{\frac{\chi_0 \omega}{2r}} \cdot e^{-j(\omega t - kr - \pi/4)} \qquad (2)$$

$$p_s = j \frac{\rho_0 c k R^2}{2r} V_0 \cdot e^{j(\omega t - kr)} \left[\frac{2 J_1(kR \sin\theta)}{kR \sin\theta} \right] \qquad (3)$$

where ρ_0 is the absolute air density under standard conditions, R is the radius of the pitch circle of the wheels, $R = mz/2$, V_0 is the oscillation velocity amplitude, r is the distance of the point P from the gearing pole, ω is the pulsation of the noise-generating vibrations and k is ω/c; for R, r and θ see Fig. 1.

The acoustic intensities of the two types of waves are obtained with the aid of the known relation: /137

$$I = \frac{p^2}{2 \rho_0 c} \qquad (4)$$

where $\rho_0{}^0$ is the air's specific acoustic impedance,

$$\rho_0{}^0 = 414 \text{ Ns/m}^3.$$

The acoustic pressure level L between any point P of the acoustic field is the result of summing over the acoustic intensities according to the relation:

$$L = 10 \lg \frac{I_c + I_s}{I_0} \text{ [dB]} \qquad (5)$$

where I_0 is the reference intensity, $I_0 = 10^{-12}$ W/m^2.

The validity of Eqs. (2), (3) and (5) was verified experimentally with the aid of a special stand [5] under the conditions of a free acoustic field, in an anechoic chamber of the Timisoara "Traian Vuia" Polytechnic Institute.

The toothed wheels and the experimental conditions have the following characteristics: $m = 3$ mm, $z_1 = z_2 = 39$ teeth,

$\xi_1 = \xi_2 = 0$, $\theta = 0^o$ and 90^o, $\phi\epsilon(0.180^o)$, $\omega_1 = 255\ s^{-1}$, $r = 40$ cm.

In the noise spectrum obtained by using the Bruel-Kjaer 2112 one-third octave analyzer we found in all investigated gears two components with:

-- the frequency of the contacts between the flanks $f_z = \omega_1 z/2\pi = 1{,}600$ Hz;

-- the gear's proper frequency $f_a = 5{,}000$ Hz (from calculation and the noise spectrum analysis curve).

From the spectrogram of the gear running under the said conditions, we found the following acoustic pressure levels for the two components:

$L_z = 83$ dB and $L_a = 88$ dB.

The experimentally determined vibration velocity amplitude was $V_0 = 4 \cdot 10^{-3}$ m/s.

Calculating the acoustic pressure levels of the frequency components f_a with the aid of Eqs. (2), (3), (4) and (5) we get /138
$L_{cil} = 97$ dB, $L_{sf} = 70$ dB and $L_{tot} = 97$ dB. Since we find that $L_{tot} = L_{cil}$, the total level is not influenced by the spherical waves, which can be explained by the great difference between the two levels $\Delta L = L_{cil} - L_{sf} = 27$ dB.

The two wheels of the gear form an acoustic doublet, the radiated acoustic power is doubled and the acoustic pressure level increases by $\Delta L = 10 \lg 2 = 3$ dB.

Thus, using for calculation of the level of noise produced by the gear the model of the two wheels free in space, we get $L_a = 100$ dB as compared with $L_a = 88$ dB determined experimentally from the spectrogram with the cover of the housing being mounted.

In order to determine the influence of the housing and to separate air noise from structural (body) noise, we proceeded in the following manner:

We generated a free acoustic field with the aid of a diffuser supplied by an audio frequency generator. The output power was so regulated that the condenser microphone placed at a distance $r = 40$ cm received the acoustic pressure level $L = 100$ dB, calculated with the aid of Eq. (5). After the diffuser was introduced into the gear box, at the distance $r = 40$ cm from the gearing pole, while maintaining the generator's output power at the constant value, we measured the following noise levels ($f = f_a = 5$ kHz):

L = 74 dB, cover closed, Fig. 2 a;

L = 105 dB, with the cover of the box removed, Fig. 2 b,

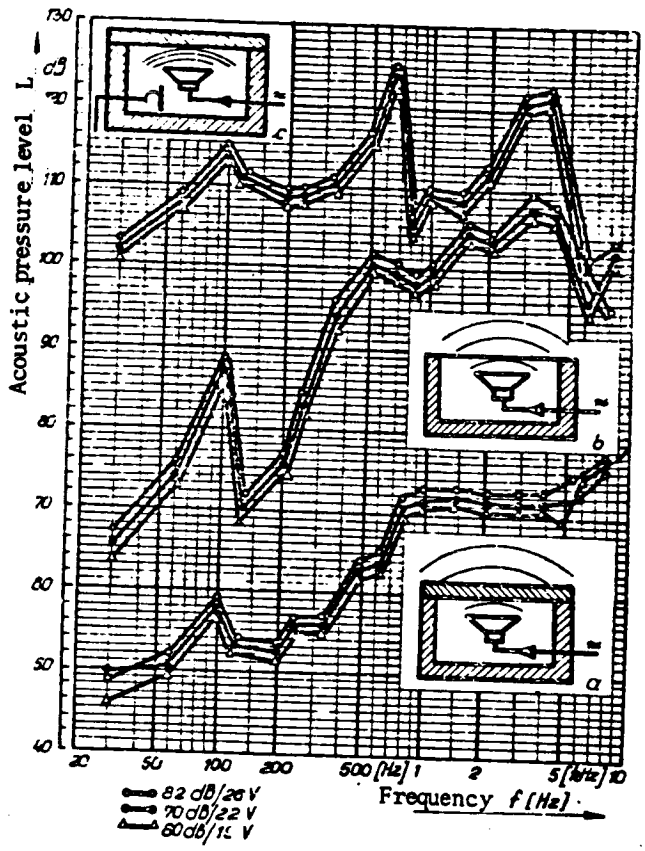

Fig. 2. Influence of the housing on acoustic radiation.

A considerable increase in the acoustic pressure level within the housing can be observed (Fig. 2 c). In this case, because of the many reflections of the sound waves, the interior of the housing behaves like a reverberation chamber, the density of the energy of the diffused acoustic field having high values.

Fig. 3 shows the attenuation of the air noise at frequencies f (30-10,000 Hz), obtained by mounting the cover of the housing, which was a 10 mm thick steel rectangular plate.

From the foregoing we can draw the following conclusions:

-- in the case of open gears there is good agreement between calculations conducted on the basis of the mechano-acoustic model shown in Fig. 1 and experimental results; /14(

-- the air noise radiated by a closed gear is negligible. The acoustic field around the housing is generated by the transmission of vibrations to the outer surfaces of the housing.

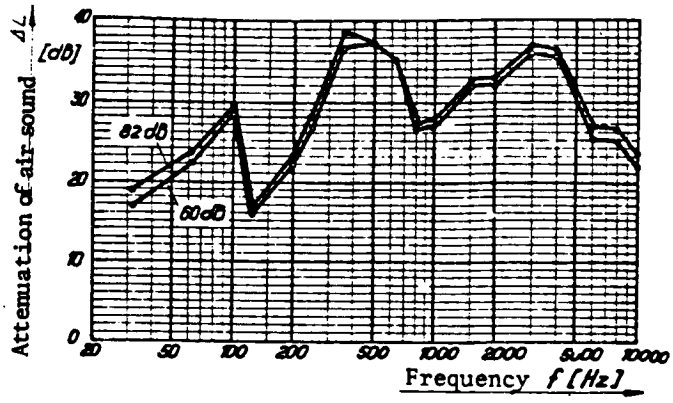

Fig. 3. Attenuation due to housing cover.

Summary

The paper describes a mechano-acoustic model for the calculation of the acoustic energy radiated by a working gear. According to this model, a gear is an acoustic coublet formed of the two wheels. The wheel teeth generate cylindrical acoustic waves while the front surfaces of the teeth behave like vibrating pistons. /141

Theoretical results were checked experimentally and good agreement was obtained with open gears. The experiments have shown that the air noise effect is negligible as compared with the structural noise transmitted to the gear box.

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PRELIMINARY STUDY OF GEAR NOISE CHARACTERISTICS

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1. Introduction

Today, the quality and performance of a gear is defined not only by its lift but also by its noise, vibrations, kinematic precision, processing and cheap maintenance, which are interdependent parameters. /143

A general observation in this connection is that, even in the case of low powers (~3 Kw), gears constitute a significant source of noise and high noise levels (89 dB) ought to be expected from them.

Studies aimed at quantitative and qualitative elucidation of the causes of gear noise and at reducing it or interpreting its characteristics are, therefore, particularly useful, and the present study belongs to this category.

Taking into account the fact that the total noise level has been the object of many previous studies, in what follows we shall understand by "noise characteristics" the data that can be obtained from analysis of the frequency spectrogram.

2. Laboratory Phenomena in Gears

Vibratory phenomena in gears are due to the following:

a) The gearing force, which is variable in time:

$$F(t) = \frac{a_{0f}}{2} + \sum_{1}^{\infty} (a_{nf} \cos n\omega_c t + b_{nf} \sin n\omega_c t) \qquad (1)$$

in which: $\omega_c = z\omega$, z is the number of teeth and ω is the angular velocity, $a_{0f} \ldots a_{nf}$, $b_{1f} \ldots b_{nf}$ are the coefficients of the Fourier series, calculated as a function of the number of teeth z, the mode and manner of inclination of the teeth, the velocity ω, the transmission ratio, the gearing force F, the modulus of elasticity and the pitch errors [3, 7, 8].

b) The deviations from correct shape and position of the flanks of the teeth, which come into play as a perturbing factor [3]; among these, of special interest are the profile errors, which can be expressed as follows:

$$e(t) = \frac{a_{0e}}{2} + \sum_{1}^{\infty} (a_{ne}\cos n\omega_e t + b_{ne}\sin n\omega_e t) \qquad (2)$$

in which: $\omega_e = \lambda \cdot \omega_c$, λ is a coefficient (according to ref. [7] it is equal to unity), $a_{0e} \ldots a_{ne}$, $b_{1e} \ldots b_{ne}$ are the coefficients of the Fourier series calculated as a function of the geometrical characteristics of the profile.

c) Active and passive shocks against the flanks in the case of temporary losses of contact; for normal research situations such losses of contact are not very probable [3]. /144

d) Rapid evacuation of air or lubricant from between the teeth during gearing; for normal rates of speed the effects of this phenomenon are slight [6].

The above-mentioned phenomena can excite proper vibrations in the reducer for teeth, toothed wheels, shafts and housing, depending on design, dimensional and mounting characteristics.

The importance of the role played by any one of the above-mentioned causes of vibratory phenomena, therefore, depends on the operating, dimensional, design and mounting characteristics of the gear under consideration.

The vibrations of toothed wheels located on shafts and of the housing on the outside under the action of variable forces in the bearing generates sound waves.

Under these conditions, in the sound and vibration spectra of a gear the following significant frequencies can be observed:

-- frequencies $n\omega_c$
-- frequencies $n\lambda\omega_c$ | position in spectrogram varies with ω_c,

-- excited proper frequencies -- relatively invariable position in spectrogram.

With a view to reducing the vibrations and noise produced by gears, the designer has at his disposal:

-- replacement of materials, if their mechanical resistance is not a determinant factor;

-- selection of the most favorable, from the point of view of noise and vibrations, complex of design factors: module, number of teeth, inclination;

-- improvement of machining with an increase in precision, or with adequate changes in the profile shape;

-- noise and vibration insulation in the reducer housing or in the outer boxes.

The first two solutions, at no additional cost, lead to reductions in the noise level by 3...6 dB. More precise machining is more efficient but at the same time much more expensive.

The efficiency of insulating boxes according to data in the literature [4, 5], can also be estimated on the basis of the results of the present study.

3. Experimental Part

3.1. Description of Setup

a. The experimental setup consisted of: a motor with from 600 to 2,800 rpm; 127 V, 2.2 Kw; an AIRBORNE ANGL gear reducer with conical toothed wheels with straight teeth and a housing of AL alloy, i = 26/13, 215 Kw, m = 2.08 mm, tooth width 11.45 mm, 6.55 Kg, equipped with slide bearings; generator with I_{max} = 11.6 A, 1,300 rpm; a discharging resistor.

The entire setup was mounted on a cast-iron foundation plate in a 3.54 x 3.82 x 4.95 m housing that was not treated phonically.

The moments of intertia of the motor and generator were decoupled by means of flexible couplings with intermediate elements made of rubber.

b. The insulating boxes utilized for all component parts /145 of the experimental setup were manufactured from 25.4 mm fiber glass plus 3.2 mm asbestos sheet plus 32 mm mineral fiber plus 3.2 mm asbestos sheet (from the inside out), with 22 Kg/m^2 and a bending strength of B = 34.4 Nm. The reducer's insulating box, which was subjected to acoustic measurements, had the following internal dimensions: 0.483 x 0.635 x 0.990 m.

c. We used a complex measuring and recording apparatus manufactured by Brüel & Kjaer, Tektronix, Federal Scientific, Moseley, Interdata, and IBM, hooked up in different ways, depending on the purpose (Fig. 1).

3.2. Experimental Results

a. The acoustic measurements on the insulating box were conducted under two circumstances:

-- box closed with five walls, suspended by means of a cable; we obtained data about attenuation (toward the inside, toward the

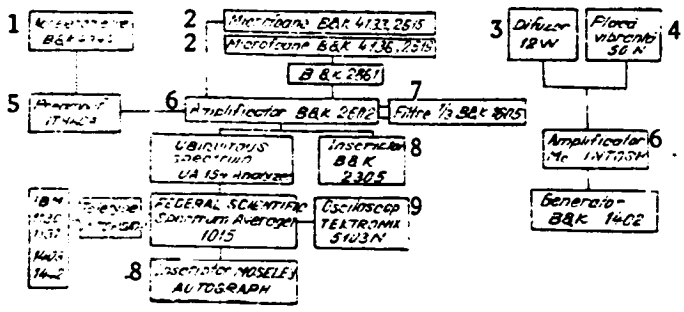

Fig. 1. Block diagram of devices utilized. 1. Accelerometers; 2. Microphones; 3. Diffuser; 4. Vibrating plate; 5. Preamplifier; 6. Amplifier; 7. Filters; 8. Recorder; 9. Oscilloscope.

outside) (Fig. 2a, b), the loss factor for the inner space, walls, chamber (Fig. 2c), the loss through transmission (Fig. 2d);

-- the box closed below and situated on a 25.4 mm fiber glass layer, with two holes for shafts in the walls (Ø 25.4 mm); we obtained data about attenuation (toward the inside, toward the outside) (Fig. 2a,b).

The microphones were placed in three positions within the box and in six positions in the chamber. Five accelerometers were placed in five different positions on the wall of the box. Results were obtained by averaging.

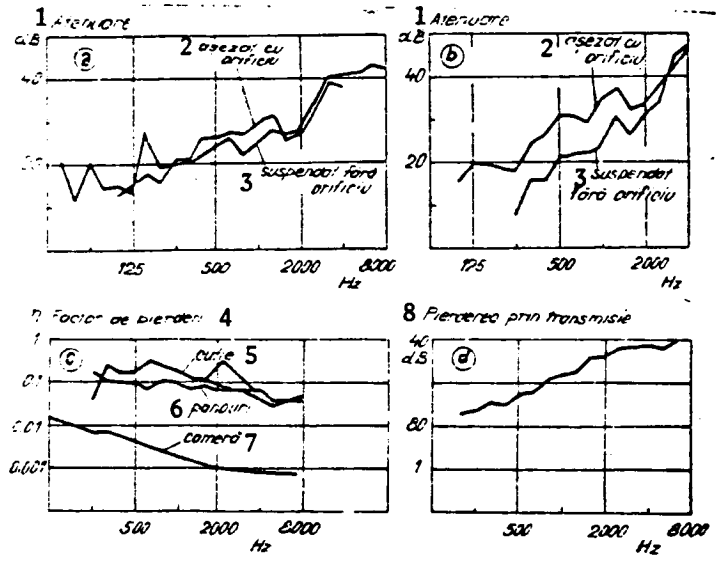

Fig. 2. Results of acoustic measurements on the insulating box. 1. Attenuation; 2. Placed with orifice; 3. Suspended without orifice; 4. Loss factor; 5. Box; 6. Panels; 7. Chamber; 8. Loss through transmission.

b. Recordings of the noise and vibration spectra for the reducer with toothed wheels were conducted for four different speeds (900, 1,200, 1,600, 2,600 rpm) and four different loads ranging from 0 to 1,630 N. The results obtained by comparison of spectra of the type shown in Fig. 3a are shown in Fig. 3b with the number of peaks observed; Fig. 3c shows the influence of the load on the peak level; Fig. 3d shows the influence of the speed and Fig. 3e shows the position of the peaks as a function of load and speed.

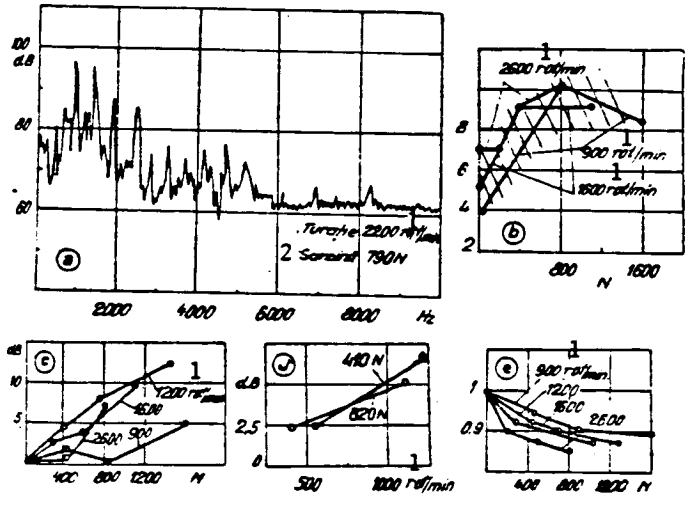

Fig. 3. Reducer noise and vibration measurement results. Examples and comparisons. 1. rpm; 2. Load.

The microphones were placed inside the reducer at 20 mm from the gearing pole along the direction normal to the contact, outside the reducer at 50 mm from the housing in the direction normal to the contact, idem, along the extension of the pinion's axle. The accelerometer was placed on the housing of the reducer in the direction normal to the contact. /146

We also placed microphones for checking and comparison inside the insulating boxes for the motor and the generator brakes these accelerometers on the base of the motor and the support of the reducer.

4. Conclusions

a. With all their reduced weight, the insulating boxes used proved to be efficient, especially at the frequencies that are

significant for gears (attenuation > 25 dB at frequencies ≥ 1,000 Hz); application of statistical energy analysis (SEA) led to results comparable to those obtained only in the experimental way (differences for losses through transmission ±3 dB).

b. Study of gear noise and vibration spectra with the aid of narrow-band (10...20 Hz) analysis with averaging over 256 spectra led to much more significant results than in the case of analysis with the 1/3 octave band.

c. The noise and vibration spectra showed aspects similar to the peaks (90...98 dB re 2 x 10^{-5} N/m^2, or -14...-8 dB re 1 g for 500...2,000 Hz) corresponding to the multiples of the contact frequencies or the frequencies of the excited vibration of the torsion system in the assembly, with possible influences of the mounting (Fig. 3a). /147

d. The coincidence of peak position with multiples of the contact frequency or with the frequencies of the excited (proper) vibration were obtained by decoupling or coupling additional shafts in the system.

e. The number of peaks observed (with differences > 10 dB) increases with the load (Fig. 3b).

f. For measurements conducted with the microphone inside the reducer box, an increase in the load led to an increase in the peak level (Fig. 3c).

g. An increase in the speed, for all experimental conditions, led to an increase in the peak level (for 75...81% of the peaks observed) (Fig. 3d).

h. The housing of the reducer had noise damping effects of about 10 dB only for the load frequencies (the first 57% of the peaks) and only for microphone positions normal to the point of contact. For lateral positions of the box damping is reduced (2.5 ... 5 dB) and occurs for a smaller number of peaks (38%). In keeping with its weight, the housing -- especially at high frequencies -- did not constitute an efficient means of damping; the vibrations generated in the gear were transmitted through the bearing to the box and the support, which became noise sources. Consideration of the reducer as a system with two masses -- toothed wheels and housing -- mounted in series on elastic elements -- shafts and fastening screws on the support -- pointed to the possibility of two resonance regions, 1,250 and 5,000 Hz, which turned out to be present in many spectra. /148

i. The variations in level for the above-mentioned peaks were considerable: differences ≥ 10 dB for frequency variations < 5%. This observation, together with those mentioned in

paragraphs d, e, f and g, justifies the present tendency to use signals on tuned frequencies or on filtered frequency bands for diagnosing or checking machine operation. Every change in experimental conditions was reflected in the spectrum as significant changes in the peak level, which acted as amplifiers for load, speed and mounting variations.

Summary

The authors present experimental results and conclusions about the noise characteristics of gears as a function of their operating conditions. The interpretive possibilities offered by real-time noise and vibration spectrum analysis are emphasized. /149

The noise reducing effects of light-weight insulation boxes are also discussed.

This study was carried out in the Acoustics and Vibrations Laboratory, Mechanical Engineering Department, MIT, under the direction of Professor Dr. Richard Lyon.

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N74-28381

LOAD INFLUENCE ON GEAR NOISE

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It is known that an increase in the transmitting load leads to an increase in the noise produced by a running gear. /151

Especially significant is the fact that an increase in load changes the influences of other parameters on the noise level.

A mathematical model proposed by the author of the present study at the Seventh International Congress on Acoustics in Budapest, 1971 [1] permitted equations to be set up for calculation of the noise of a running gear and of the influences of some parameters on it.

For the spectral component corresponding to the tooth contact frequency the acoustic pressure level is determined by means of Eq. (1) adopted to the FORTRAN language:

(1)

The equation for calculating the spectral component corresponding to the gear's proper frequency is shown in the organogram of Fig. 1.

In the equations, the following notations are used:

P is the pulsation of the gear's forced vibration (1,117... 12,461 s^{-1});

D is the width of the gear rim ($33 \cdot 10^{-3}$);

A is the distance between the axes ($115 \cdot 10^{-3}$); /152

G is the modulus ($(2.5, 5) \cdot 10^{-3}$);

E is the reduced error of the tooth shape ($(23, 30, 39) \cdot 10^{-6}$ m);

H is the viscous damping constant (1,525 Ns/m);

V is the gear's rigidity ($6.266 \cdot 10^{8}$ N/m); /153

X is the reduced initial phase (0.832 radians); /154

Q is the reduced mass of the gear (0.565 Kg);

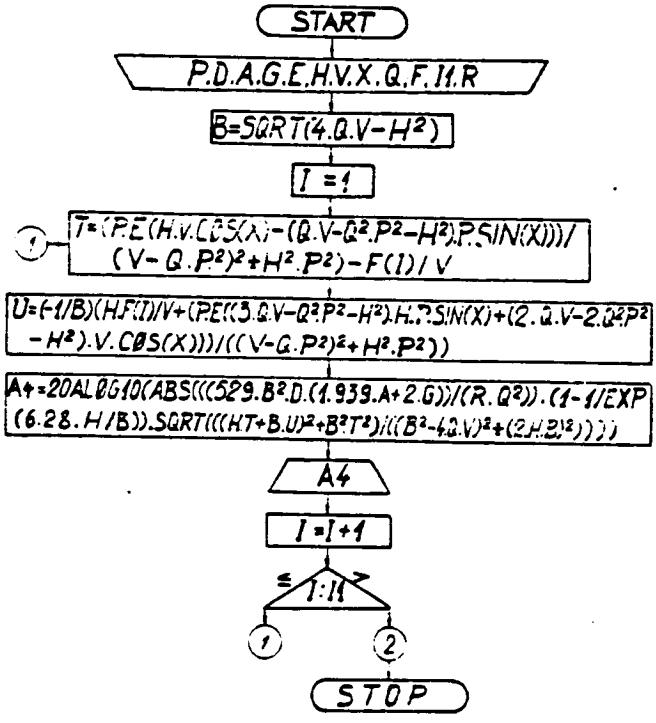

Fig. 1.

F is the normal force on the teeth; /15

I1 is the number of values of the force F adopted in the study; /15

R is the distance from the gearing place to the point of the surrounding space in which the acoustic pressure level of Eq.(1) is determined;

A4 is the acoustic pressure level (dB).

The curves of Figs. 2...7 show the calculation as well as the experimental results concerning the reduction in influence of the speed and of the error of the tooth profile shape owing to the increase in load. Analysis of these curves leads to the following results:

-- with increasing load the influence of the speed (Fig. 2) and of the error of the tooth profile shape (Fig. 5) on the acoustic pressure level corresponding to the gear's proper frequency decreases;

-- an increase in load, therefore, leads to small reductions in the influence of the speed and of the error of the tooth profile shape on the acoustic pressure level corresponding to the contact frequency, as well as on the total level;

-- calculation results are close to the experimental results.

Summary

An original mathematical model proposed by the author has permitted equations to be derived for calculation of gear noise. /15

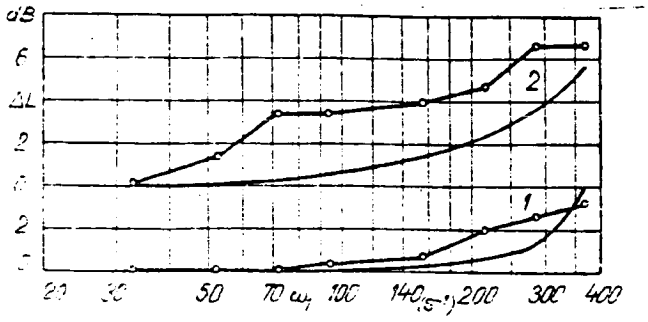

Fig. 2. Reduction ΔL(dB) of the influence of the speed on the acoustic pressure level corresponding to the gear's proper frequency, due to an increase in the load from F = 3,730 (N) to F = 7,650 (N)(curve 1) and to F = 12,700 (N) (curve 2). The continuous curve represents calculation results and the curve with circles represents experimental results.

These equations permit the acoustic pressure level to be determined as a function of load.

The author applies this method to three parallel gears.

The logical calculation schema is given, as well as the results obtained.

The values obtained by calculation are close to the experimental results.

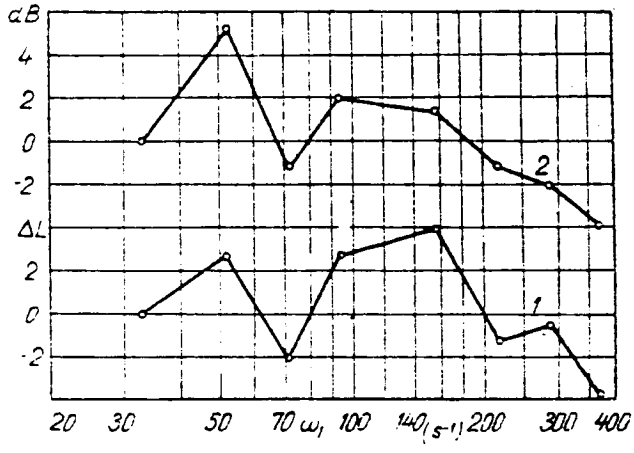

Fig. 3. Change ΔL(dB) in influence of speed on acoustic pressure level corresponding to contact frequency, due to increase in load from F = 3,730 (N) to F = 7,650 (N) (curve 1) and to F = 12,700 (N) (curve 2).

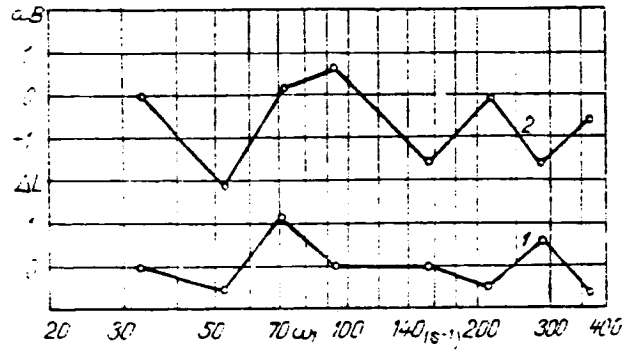

Fig. 4. Change ΔL(dB) in influence of speed on total acoustic pressure level, due to increase in load from F = 3,730 (N) to F = 7,650 (N) (curve 1) and to F = 12,700 (N) (curve 2).

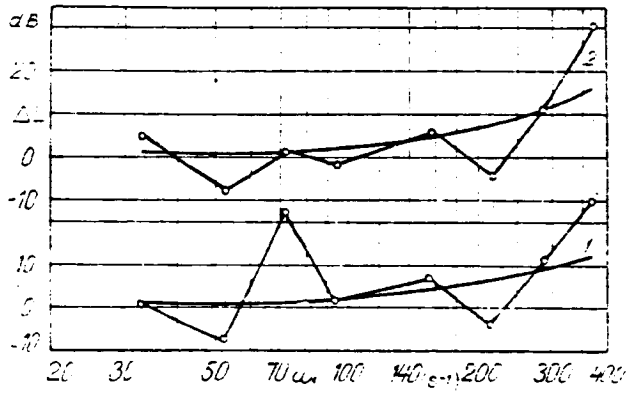

Fig. 5. Reduction ΔL(dB) in influence of error of tooth profile shape on acoustic pressure level corresponding to gear's proper frequency, due to increase in load from F = 3,730 (N) to F = 7,650 (N) (curve 1) and to F = 12,700 (N) (curve 2). The continuous curve represents calculation results and the curve with circles represents experimental results.

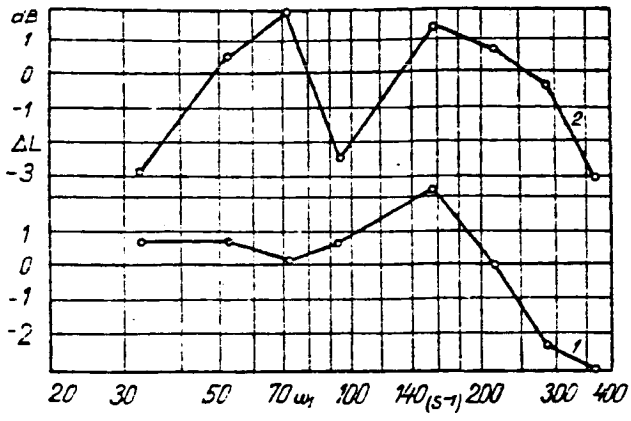

Fig. 6. Change ΔL(dB) in influence of error of tooth profile shape on acoustic pressure level corresponding to contact frequency, due to increase in load from F = 3,730 (N) to F = 7,650 (N) (curve 1) and to F = 12,700 (N) (curve 2).

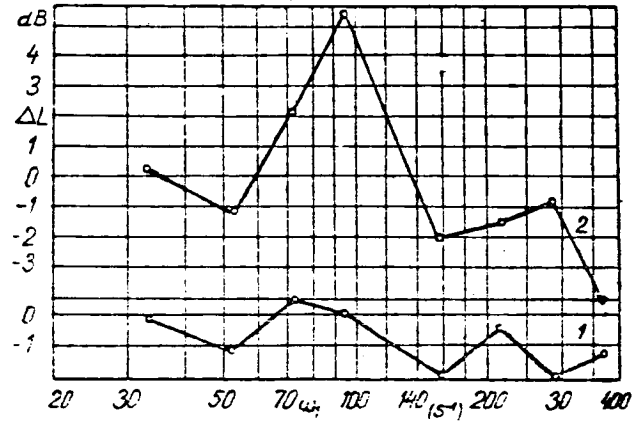

Fig. 7. Change ΔL(dB) in influence of error of tooth profile shape on total acoustic pressure level, due to increase in load from F = 3,730 (N) to F = 7,650 (N) (curve 1) and to F = 12,700 (N) (curve 2).

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N74-28382

CONTRIBUTIONS TO THE REDUCTION OF GEAR NOISE

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In the specialty literature, the influence of profile displacements of a noise produced by toothed gears is mentioned only in passing, in connection with the study of the influence of the number of teeth, the modulus, the extent of covering, etc. [5, 6]. /15

At the acoustical research laboratory of the Department of Machine Parts and Mechanisms of the "Traian Vuia" Polytechnic Institute of Timisoara, systematic studies were conducted of the silentness of gears obtained by means of about 80 combinations of toothed wheels having the following profile displacements: $\xi_{12} = 0; \pm 0.2; \pm 0.4; \pm 0.6; \pm 0.8; \pm 1$ and ± 1.2.

The gears were studied on a stand of the circuit type (energy flux) closed mechanically, for the specific load $Q/B = 5; 30; 50$ N/mm and the angular velocity $\omega_1 \varepsilon (50...250\ s^{-1})$.

Figs. 1...4 show the spatial distributions of the noise levels of running gears on a semi-spherical surface having its center in the pitch point. The radius of the sphere R_s is much greater than that of the pitch circle $R(R_s = 400$ mm, $R = 58$ mm).

Measurements were conducted in the frontal plane of the wheels (Figs. 1...4), as well as at different angles $\alpha \varepsilon (0, 180^o)$ with respect to the horizontal plane. The condenser microphone was fastened at the angles $\beta \varepsilon (0, 180^o)$ with respect to the center line.

Figs. 5 and 6 show the variation in the acoustic pressure level L as a function of the angular velocity for certain specific loads Q/B.

Analyzing these representative curves, we arrive at the following results: /16

-- a directivity of the radiation of acoustic energy can be observed along and perpendicular to the gearing line, the increase in the level along these directions being 2-4 dB, which agrees with the results obtained by other researchers [1, 4];

-- when the cover of the housing (a rectangular steel plate, thickness $\delta = 10$ mm) is removed, the acoustic pressure level drops by 10-16 dB, Fig. 4; /16

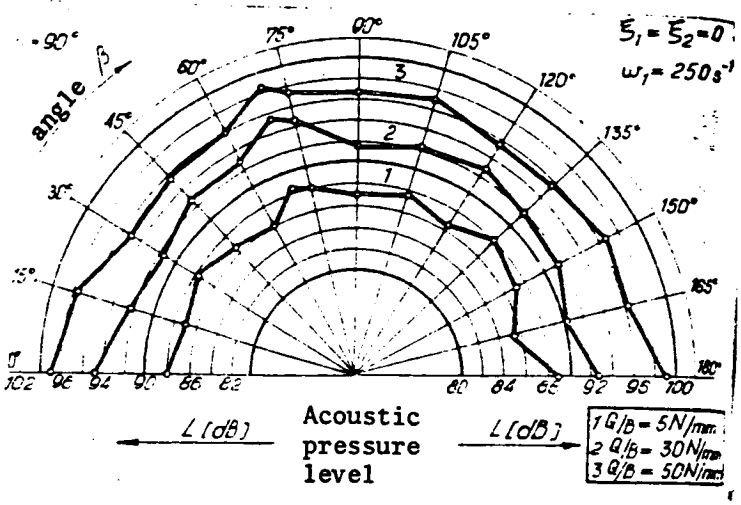

Fig. 1. Spatial distribution of noise level for the zero gear, $z_1 = z_2 = 39$, m = 3 mm.

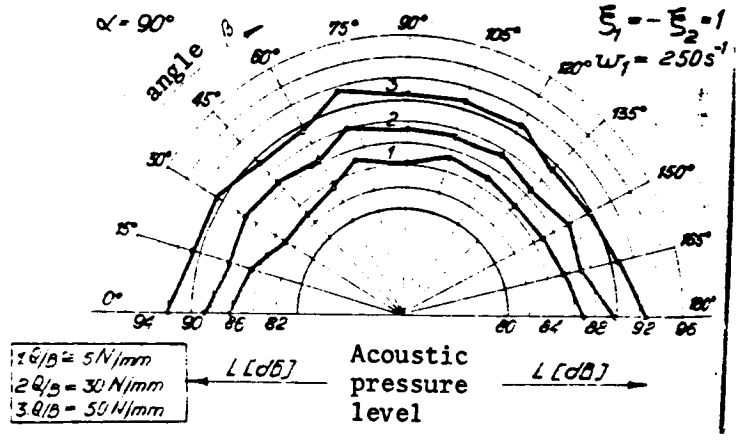

Fig. 2. Variation in the acoustic pressure level L of the post-polar gear, $z_1 = z_2 = 39$, m = 3 mm.

-- when the specific load Q/B is kept at a constant value and the values of the angular velocity are doubled, the noise level increases by 5-6 dB, Fig. 5 and Fig. 6;

-- when the angular velocity is kept unchanged and the specific load is doubled, the total noise level of the gear increases by 3-4 dB (Fig. 5, Fig. 6);

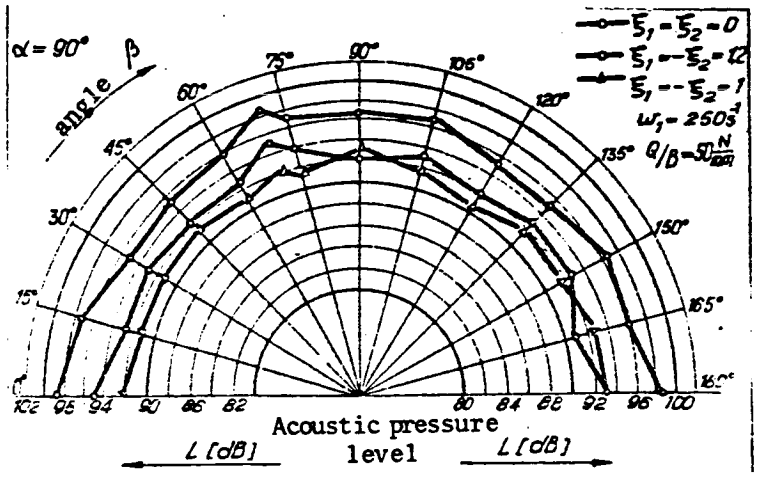

Fig. 3. Reduction in the noise level of extra-polar gears as compared with zero gears.

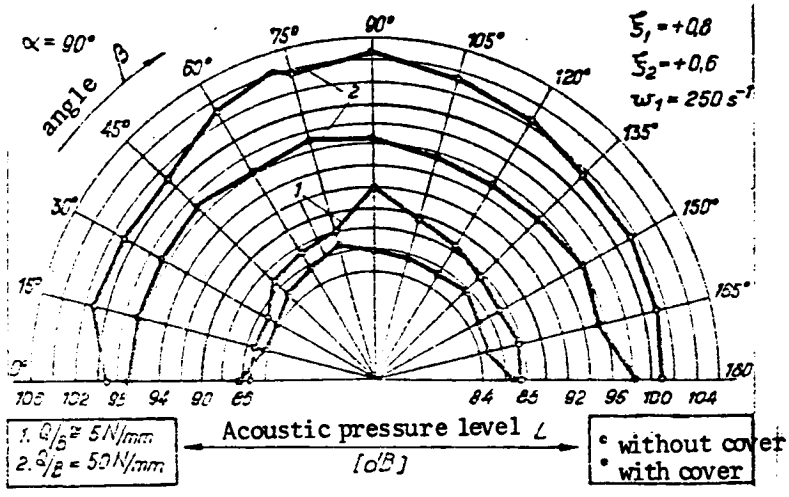

Fig. 4. Influence of the box cover on the spatial distribution of the total noise level of gears.

-- in the noise spectrum obtained with the aid of a one-third octave analyzer, two components are found to predominate in the spectrum: a component having the tooth gearing frequency

$f_a = \omega_1 z_1/2\pi$ and another having the proper oscillation frequency of the gear f_p = 5 khz [2].

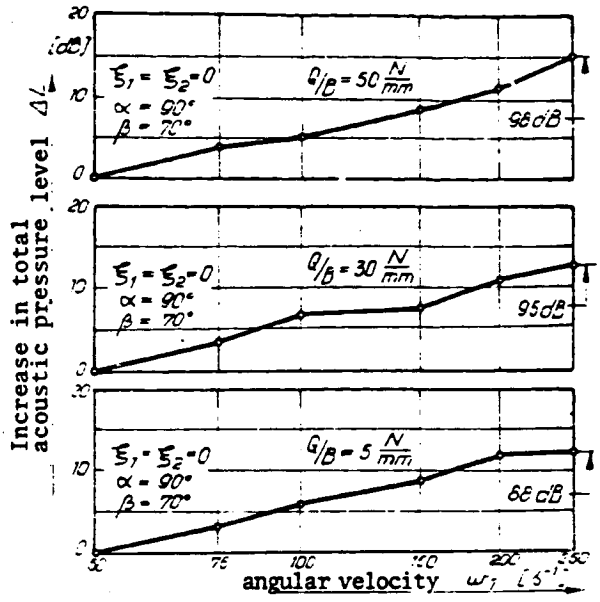

Fig. 5. Increase in total acoustic pressure level ΔL (dB) ($z_1 = z_2 = 39$, $\xi_1 = \xi_2 = 0$, m = 3 mm).

The above observations hold for zero gears as well as for displacement gears.

Studying the influence of profile displacements on the level of noise generated by gears, we can draw the following conclusions:

-- there exists a slight tendency toward noise reduction in positively displaced gears, especially when the specific displacements ξ_1 and ξ_2 have close absolute values $\Delta L = -(0.3 \ldots 0.7)$ dB;

-- in extra-polar gears, especially post-polar ones, the noise level is lower, on an average, by 3 dB (or, in the case of the intensity, by 10-12 sones) as compared with the noise generated by zero gears;

-- extra-polar gears also function well from the point of view of vibration generation;

-- comparing the spectrograms of normal gears with those of extra-polar gears, we find that in the spectrum of the first type of gear the components with fε(600-6,000 Hz) (for which the ear has a heightened sensitivity) have higher levels and fluctuations.

The experimental data obtained by objective measurements with the aid of a precision sonometer were processed with due regard for the subjective sensation produced by the noise of the gear according to the method worked out by Zwicker [7]. By using post-polar gears the intensity level is reduced by two phones.

Summary

The paper presents the results of experimental research on the influence of profile displacements on the acoustic pressure level of running gears.

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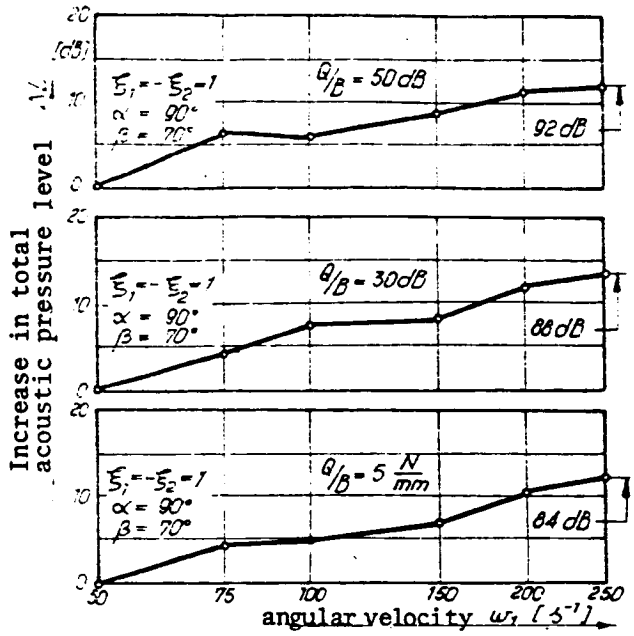

Fig. 6. Increase in the total acoustic pressure level ΔL (dB) in extra-polar gears ($z_1 = z_2 = 39$, $\xi_1 = -\xi_2 = 1$, m = 3 mm).

The conclusion is reached that extra-polar gears function well, reducing the acoustic pressure level by about 2-3 dB which, at the level of the total noise of 80-90 dB, can be easily felt.

This result can be explained by the fact that in this gear the pitch point is situated outside of the gearing segment, so that a certain change in the direction of the friction force between the tooth flanks is eliminated.

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ASPECTS OF REDUCING SPUR GEAR NOISE

Horia Giurgiuman-Negrea*, Mircea Cretu*, and Ion Moraru**

In machines where the transmission of motion is effected by means of gears the share of the noise produced by the latter in the machine's total noise is particularly important. Under these conditions it is necessary to investigate the noise sources of gears, as well as ways to improve their silentness. /16

Fig. 1 shows schematically the way that different factors, beginning with the designing and working out of the technology, continuing by way of execution and ending with effective operating conditions, determine gear noise.

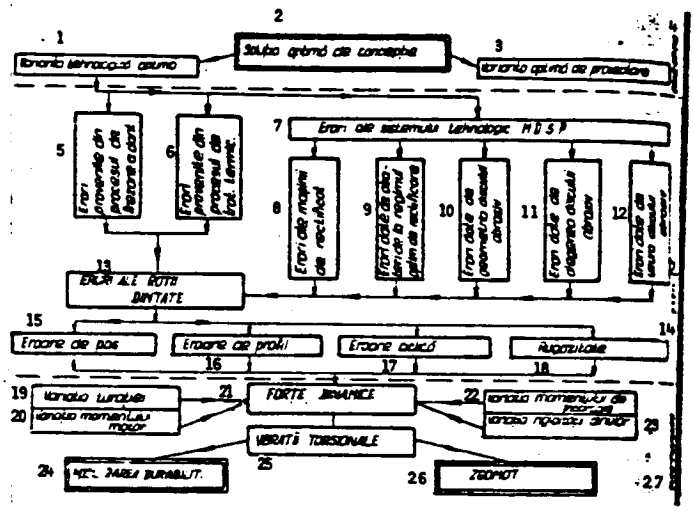

Fig. 1. 1. Optimum technological version; 2. Optimum conceptual solution; 3. Optimum design version; 4. Conception; 5. Tooth milling errors; 6. Heat treatment errors; 7. Errors of the MDSP technological system; 8. Grinding machine errors; 9. Errors due to deviations from optimum rectification conditions; 10. Errors due to abrasive disc geometry; 11. Errors due to abrasive disc choice; 12. Errors due to abrasive disc wear; 13. Toothed wheel errors; 14. Execution; 15. Pitch errors; 16. Profile errors; 17. Cyclic errors; 18. Roughness; 19. Variation in number of rotations; 20. Variation in motor moment; 21. Dynamic forces; 22. Variation

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Fig. 1 key cont'd. in loading moment; 23. Variation in tooth rigidity; 24. Reduced durability; 25. Torsional vibrations; 26. Noise; 27. Operation.

Noise sources can be classified into external sources -- variation in the number of rotations and in the transmitted moment -- and internal ones, consisting in errors of the teeth, variation in their rigidity, roughness, etc.

The research conducted at the Department of Machine Building Technology of the Cluj Polytechnic Institute dealt with the second category of noise sources.

A gear's dynamic behavior is to a great extent determined by its degree of covering ε in the sense that a high degree of covering improves the gear's silentness.

(1)

(Rb_1, Rb_2 are the radii of the wheels' basic circles, αe_1, αe_2 are the angles of pressure against the outer diameter, P_b is the basic pitch and αr is the operating meshing angle).

In the event that too low an operating meshing angle αr is chosen for a negatively corrected gear the degree of covering increases, as can be seen from the diagram of Fig. 2. /168

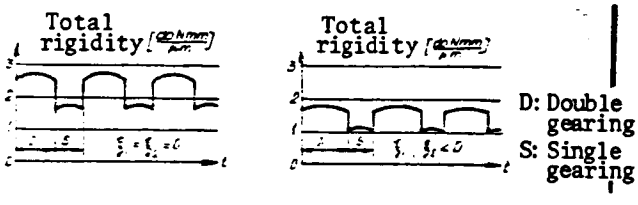


Fig. 2.

The effect of negative correction, under the conditions of the gear's existence, is not only a reduction in the difference of rigidity between the regions of double and simple gearing, but also a reduction in total rigidity, which diminishes the bearing capacity and also leads to tooth deformation, so that correction must be applied with discrimination. /169

Tooth errors -- a significant source of gear excitation -- can lead to different dynamic loads, decisively influencing functional behavior. While the cyclic error can be neglected in many cases because of the load frequencies at which it comes into play, the basic pitch error, the profile shape error and the magnitude of roughness influence the noise level significantly.

Basic pitch deviation causes the wheels to accelerate and decelerate when passing from double to simple gearing as a result of positive or negative deviations (Fig. 3), which leads to irregular running of the gear. The factor on which the noise level depends is the difference in the basic pitch deviations of the two wheels. Fig. 4 shows this dependence, the difference being considered positive when the basic pitch of the leading wheel is greater [3]. In accordance with the foregoing, either execution within extremely close tolerances or selective assembling of the function of the effective deviations would be necessary. /17

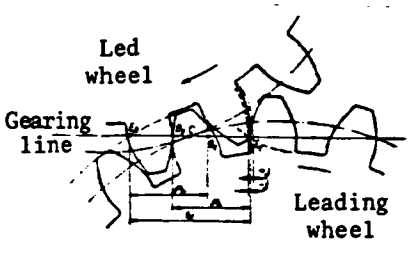


Fig. 3.

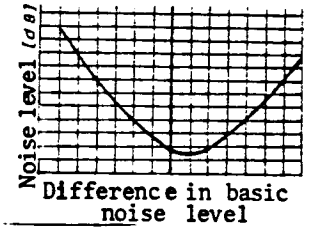
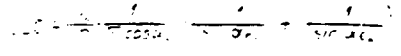


Fig. 4

The effect of the errors of profile shape and of tooth surface roughness is the appearance of high noise levels at high frequencies, even in the case of low peripheral speeds, so that it is desirable for the wheels to be qualitatively as good as possible from this standpoint.

Still from the technological point of view, for quieter running the tooth shape can be corrected by using as profile two (head flanking or foot flanking) or three connected involutes (head flanking and foot flanking) at the connecting point, which changes the angle of pressure (Fig. 5) and damps shocks when the gears are engaged and disengaged.

Because decreasing the active portion of the profile involute diminishes the degree of covering, this method can be applied within certain limits as a function of the drop $\Delta\varepsilon$ in the degree of covering [1].



(2)

(h_f is the flanking length; m is the modulus; and α_0 is the meshing angle).

Fig. 5

In view of the foregoing, we investigated, from the standpoint of improving its silence, a textile machine having transmissions by means of gears with straight and inclined teeth.

Fig. 6 shows the machine's kinematic diagram, in which the number of teeth as well as the number of rotations incident to engaging and disengaging are indicated. With the aid of a 2111 Brüel & Kjaer sonometer, we analyzed the machine's noise level on portions of the kinematic chain in order to be able to determine the contribution of each gear or part of the machine to the total noise. The results of these measurements are shown in /17
Fig. 7, which gives the noise cuves of the parts of the kinematic chain indicated in Fig. 6. The results were confirmed by measurements conducted on a PRI-320 Daldi & Matteucci noise and vibration stand. In this fashion we were able to identify the gear with the highest noise level, which has the following characteristics: $z_1 = 98$; $z_2 = 65$; $m_n = 2$; $\beta_0 = 15^o$. On this gear we tested some methods of improving the noise level, going on to apply them also to other gears that are silent but little.

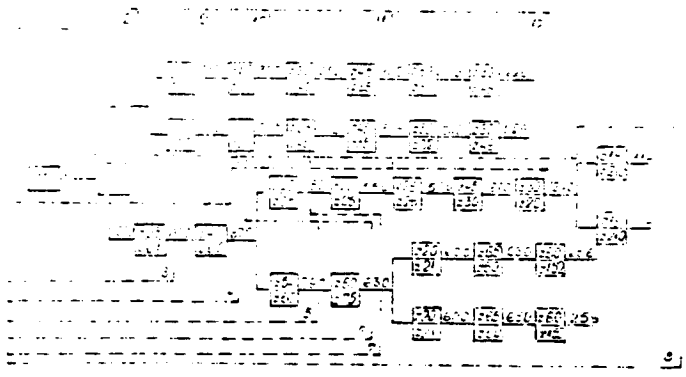


Fig. 6.

Because the basic pitch deviations on the periphery of the wheels are strictly uniform, the difference in deviations is constant during operation. Noise level measurements agree with those obtained previously with regard to the influence of the /17
basic pitch and are shown in Table 1.

TABLE I

Difference in basic pitch deviations	Frequency	Line
	Noise level	

Applying a flanking to the tooth head also led to a reduction in the noise level. By testing different flankings we determined that a good head flanking is one that has the length h_f = 1.5 mm and the depth g_f = 0.02 mm. Table 2 shows the results as compared with unflanked teeth, the difference in basic pitch deviations being zero in both cases.

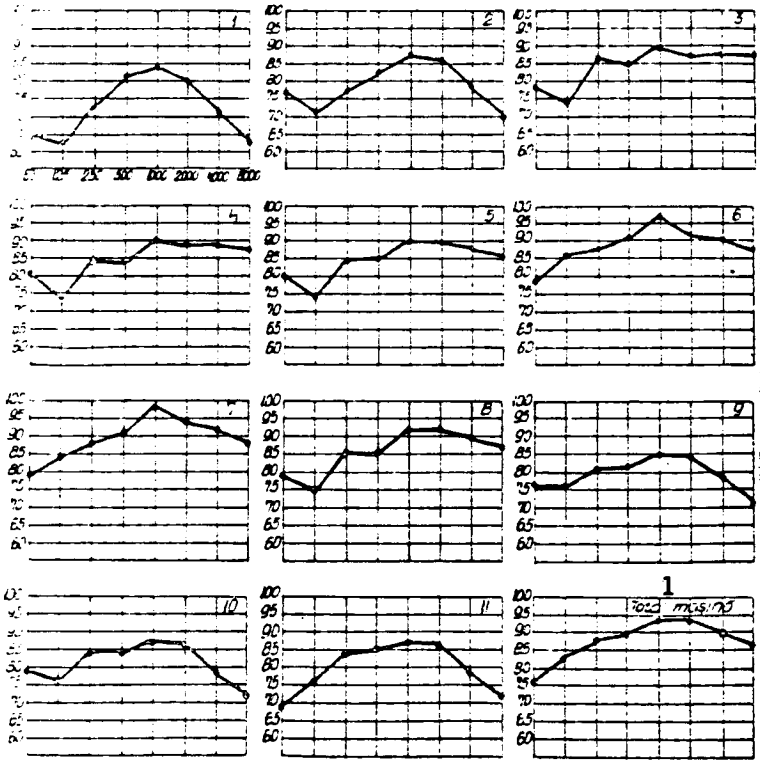


Fig. 7. 1. Machine total.

TABLE 2

Tooth pipe	Frequency (Hz)								Li
	Noise level (dB)								
Unflanked	59	60	61	60	72	65	48	45	
Flanked	52	59	61	75	72	63	56	46	

Measurements of the error of profile shape and of the roughness of the wheels we studied, which were rectified on the machine with abrasive stone, yielded very low and constant values for the entire lot (E_{fp} = 3...[illegible] and R_s = 0.8 μm), which means that no sign of their influence on silence could be found. /17

By adding a tooth to the large wheel, in the event the axial distance is maintained, the gear must be corrected negatively, having specific displacements ξ_{f1} = -0.28396; ξ_{f2} = -0.186439. The operating meshing angle becomes α_{rf} = 19°41'40" as compared with α_{of} = 20°38'50". This corresponds to a substantial increase in the degree of covering ($\Delta\varepsilon$ = +0.90338), which can lead to an enhancement of silence by 3-4 dB.

In the present case too, by the measurements indicated for improving the quiet running of a gear, characteristics can be obtained that are satisfactory from the point of view of the demands of modern machine building.

Summary

This paper is a contribution to industrial noise reduction. /17
From this point of view the authors discuss some noise sources in gears and ways of improving their silence.

Finally, the authors discuss some practical results obtained from the noise point of view during the study of a textile machine.

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N74-28384

CONTRIBUTIONS TO THE STUDY OF THE CHARACTERISTICS OF BEARING NOISE AND VIBRATIONS

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1. Introduction

Analysis of the causes and characteristics of the noise and vibrations generated by running bearings constitutes one of the principal lines of research aimed at improving the quality of these widespread machine parts. Many cases could be cited of electromechanical, cinematographic and office machines, among others, in which the bearings, as compared with other possible sources of noise and vibrations, are of preponderant importance with serious effects on the quality of the assembly, so that further study of this matter needs no justifictaion. /175

The purpose of the present study is to present some of the results obtained by scientific collaboration between the Iasi Polytechnic Institute and the Birlad Bearing Factory.

2. Noise and Vibration Sources in Running Bearings

Earlier studies[1...7] permit the following general conclusions to be drawn:

a. Between the noise and vibrations of a separate bearing and the noise and vibrations of an assembly in which the bearing in question is the single source, there exists an interdependence.

b. Between the vibrations and noise of a bearing or of an assembly with a mounted bearing there exist qualitative dependences.

c. When a gear is running, the complex relative motion of the rings on the rolling bodies can constitute a source of forced vibrations as a result of the bearing clearance and deformations and of deviations from the correct form (magnogeometric, undulatory, microgeometric); for similar reasons, the motion of the cage can also constitute a source of vibrations. /176

d. Any bearing element (rings, rolling bodies, cages) or assembly element can constitute a source of excited vibrations.

e. Like other factors that define the quality of bearings, noise and vibrations depend on the design, material, construction, mounting and operating conditions.

f. The noise and vibration level-frequency spectrogram constitutes a very sensitive characteristic that provides information about the principal sources, the factors that influence them, the appearance and evolution of deterioration phenomena, etc.

Consequently, for the proposed research stage, our experimental investigations were aimed at:

-- elucidation of some specific aspects of the noise spectrograms for bearings with the same standard dimensions;

-- analysis of the noise-vibration interdependence for a free bearing;

-- analysis of the influence of the cage material, number of rotations per minute and load on the characteristics of the noise and vibration-frequency spectrogram.

3. Experimental results

3.1. Research Conditions

a. The research chamber (3.20 x 1.90 x 2.40 m) of the Department of Machine Parts is of the anechoic type with antireflectant treatment with SILAN mineral wadding prisms (100 x 100 x 200 mm) on a 50 mm thick mineral wadding support.

Other characteristics: total absorbent surface/wall surface = 120/40 m^2; coefficient of reflection = 0.4/50 Hz/0.1/100 Hz; lower limiting frequency = 370 Hz; deviation from the law 1/r = ±0.2 dB; external noise damping = 69 dB.

b. The research stand was made-up of a supporting shaft for the bearings under study that was situated on slide bearings, with external activation of the anechoic chamber at 720, 1,440 and 2,880 rpm. The radial loads applied by the constant weight was 127, 300 and 500 N.

c. The bearings we studied were 6308 EL radial bearings with balls and NJ206 bearings with cylindrical rollers, with dimensional measurements respecting their radial and axial beat, /17
polygonality, ovality, undulation and diameters, which fell within the limits of the P5 precision class.

d. The noise and vibration measuring instruments were of the Brüel & Kjaer type (4131 microphone, 4334 accelerometer,

2112 spectrometer, 2305 recorder). The microphone was placed at a distance of 0.5 m in correlation with the measured directivity characteristic. The accelerometer was likewise placed in the direction of maximum radial acceleration of the outer ring of the bearings.

3.2. Experimental Results

a. Fig. 1 shows the results of statistical interpretation of the noise measurements conducted on 511 NJ206 bearings and 435 6308 EL bearings.

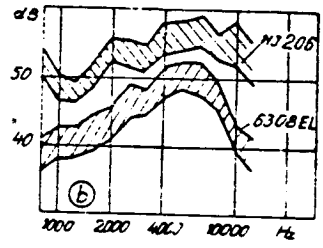


Fig. 1.

b. Fig. 2 (example) shows the noise-vibration interdependence for a 6308 EL bearing.

c. Fig. 4a, b (examples) shows comparative data on the influence of speed and load on the noise spectrograms.

d. Fig. 3 shows a noise spectrogram for a 6308 EL bearing with an ordinary metal cage and a polyamide cage.

e. The drive conditions of the bearing-holder shaft being given, the influence of the background noise was separated, taking into account the level of this noise by finally considering only frequencies greater than 800 Hz.

4. Conclusions

Analysis of the experimental results leads to the following conclusions: /17

a. In general, the measured noise spectra indicate prominent zones that are independent of the number of rpm for the frequency bands centered on 2,500 and 6,300 Hz, which coincide with the proper flexure frequencies of the outer rings; the presence of these prominent zones is even visible in the spectrograms of Figs. 2, 3 and 4.

b. A qualitative relationship can be seen between the noise and vibration spectrograms.

c. Increasing the load did not lead to appreciable changes in the spectrum. /17

d. Increasing the number of rpm led to a growth of the spectrum's high frequency zones, which was more pronounced for the frequency coinciding in position with the excited proper frequencies.

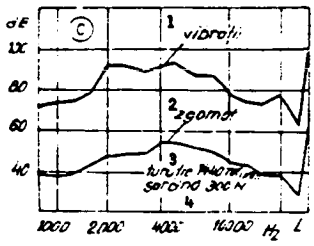


Fig. 2. 1. Vibrations; 2. Noise; 3. 1,440 rpm; 4. Load.

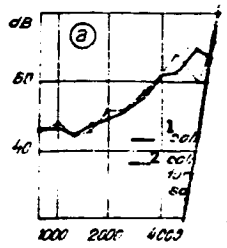
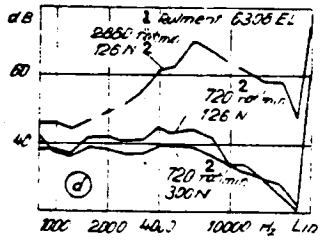
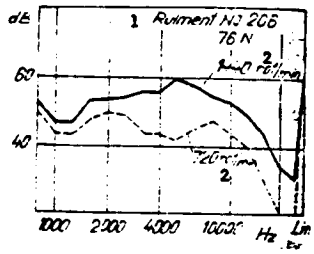


Fig. 3. 1. Steel cage; 2. Polyamide cage; 3. 2,880 rpm; 3. Load.



a



b

Fig. 4. 1. Bearing; 2. rpm.

e. Changes in the spectrum under the influence of the investigated factors turned out to be much more evident than changes in the total level.

f. Radial bearings with cylindrical rollers were less silent than radial bearings with balls (Fig. 1).

Summary

The paper presents the results of experimental research on the noise and vibrations of radial ball and roller bearings. Analysis of the frequency spectra disclosed dependences among vibration and noise, load, rpm and cage material. Some peculiar aspects of the frequency spectrum are also mentioned.

/18

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N74-28385

DATA ON THE NOISE VIBRATIONS OF MODERN TRACTION LOCOMOTIVES

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The widespread introduction of modern locomotives calls for increased attention to the noise and vibration level, especially in the driver's cabin. /18

Foreign [1] as well as domestic [2] studies show that the noise and vibrations of modern locomotives constitute an important factor of discomfort that has a negative effect on the drivers.

At the present time research is being conducted in three principal directions:

-- determination of noise and vibration sources;

-- correlation with the nature and incidence of the changes produced in the human organism;

-- solutions that are required in order to reduce these factors.

Noise and Vibration Sources

While Diesel engines constitute the principal noise and vibration source in Diesel-electric and Diesel-hydraulic traction locomotives, the electric engines of electric locomotives play a secondary role; in the latter case it is air compressors that represent the principal source.

Specialists [3] mention five principal causes of noise and vibrations in Diesel engines (Fig. 1): exhaust of burnt gases, air suction, the noise emitted by the engine's outer surface, the noise emitted by the parts mounted on the engine and the noise of auxiliary devices. In principle, a distinction can be drawn between the direct noise generated by oscillatory variations in the pressure and speed of gases in motion and the indirect noise generated by the vibrations of solid surfaces. The ratio between the energy of the sound emitted and the useful mechanical work produced by the motor increases approximately in proportion to the nominal rpm. /18

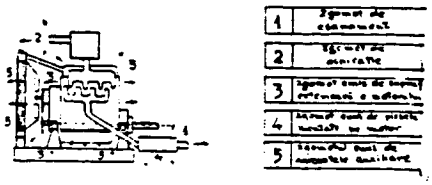

Fig. 1. 1. Exhaust noise; 2. Suction noise; 3. Noise emitted by engine's outer surface; 4. Noise emitted by parts mounted on engine; 5. Noise emitted by auxiliary devices; 6. Direct noise; 7. Indirect noise.

Intermittently superposed on this noise is the noise generated by the air compressor, which is the dominant source in electric locomotives. But in addition to these causes, on modern locomotives the causes of noise and vibrations are supplemented by the oscillatory motions of the vehicle that rolls on the rails, the underframe-bogie connections, the type of suspension chosen, the contact of the wheels with the track joints, the vibration of the carriage body walls, etc., which can lead to an adding up and amplification of the intensity of the total noise and vibration level. Nevertheless, the principal source remains the entire gearing of the engines installed on the locomotive (Fig. 2).

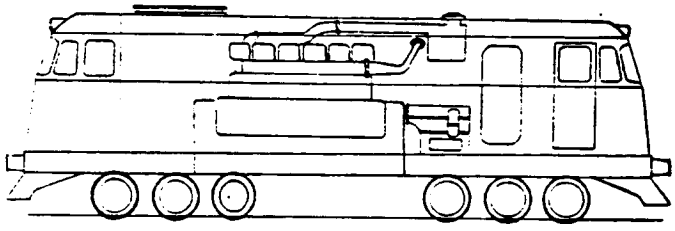

Fig. 2.

Data on the Noise and Vibration Level

The noise and vibration situation in the driver's cabin of modern locomotives is the resultant of all the sources that act during the rolling of the locomotive on the track. Measurements /18
were conducted according to the republican and international standards and the data collected were referred to the values recommended as admissible by the republican labor protection norms, the UIC and the ISO (for example, the noise curve N 75).

In the cabin of the Diesel-electric locomotives manufactured by Electroputere Craiova 2100 CP, the mean noise values were:

	Total noise	Frequency spectrum in Hz								
		63	125	250	500	1000	2000	4000	8000	16000
		93	89	85	79	67	59	55	48	36
		[illegible]	91	87	80	75	72	64	56	41
		[illegible]	94	87	82	77	73	65	57	46
		[illegible]	96	90	83	79	77	65	56	46
		115	102	97	91	84	77	73	66	59	50

The Cz 75 curve taken as limiting admissible level was exceeded as follows:

$$\Delta L = L_{measured} - L_{admissible}$$

		ΔL for measured frequencies								
		63	125	250	500	1000	2000	4000	8000	16 kHz
400	50	-	1	3	-	-	-	-	-	-
600	80	-	4	5	2	-	-	-	-	-
600	90	-	7	5	4	-	-	-	-	-
650	110	4	9	8	5	1	-	-	-	-
700	120	6	10	9	6	2	-	-	-	-

The curve was most significantly exceeded in the low-frequency region that occupies a good part of conversational frequencies. This is important because it can mask verbal communications between the driver and his assistant about the safety of operation.

Measurements conducted in the cabin of a Diesel-electric locomotive with the machine room door closed and in the machine room with the engine running with a full load yielded the following values:

Measuring site	Total level	Frequency spectrum in Hz								
		63	125	250	500	1000	2000	4000	8000	16000
Cabin with machine room door closed	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
In machine room	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

The Cz 75 curve was exceeded as follows: /18

Measuring site	Δ for spectrum frequencies								
	63	125	250	500	1000	2000	4000	8000	16000
Cabin with machine room door closed	-	16	21	15	22	20	14	13	-
In machine room	11	15	23	26	31	27	23	24	17

These measurements show that the Cz 75 curve is most significantly exceeded at high frequencies. The operation of the air compressor causes the acoustic pressure level to increase especially in the high-frequency region, where a difference of 7-9 dB can be observed as compared with the situation when the compressor is not operating.

Measurements conducted in the vertical plane on the floor of the cabin of the Diesel-electric locomotive yielded the following values:

Speed of locomotive	Frequency Hz	Acceleration g (m/s²)
60 km/h	4,3	0,0396
45 km/h	3,3	0,0345
30 km/h	3,0	0,0292
60 km/h	3,3	0,0345
45 km/h	3,1	0,0590
30 km/h	3,0	0,0630

So we find low-frequency vibration levels, vibrations that are involved in the production of fatigue and states of sleepiness in engine drivers through their action on the central nervous system and the sensory apparatus. /18

Questioning of locomotive personnel underlines the fact that noise and vibration in the command post constitute the principal cause of perturbation during work.

Audiometric examinations conducted on engine drivers on Diesel-electric locomotives before and after a run showed no signs of occupational hearing impairment, as has been stressed by other authors, too [4].

Audiograms taken of engine drivers after a run (after 3-4 hours of noise exposure in the cabin) disclosed 51.7% in whom the hearing threshold increases by 2-7 dB as compared with the situation before getting on the locomotive. The most pronounced hearing loss is found at frequencies of 500 Hz (5 dB), 1,000 Hz (6 dB), 2,000 Hz (7 dB) and 4,000 Hz (3 dB). We consider that this temporary rise in the hearing threshold is due to fatigue of the organ of hearing, a fatigue from which a complete recovery is effected from one day to the next.

Thus, we find a causal relationship between the level of noise in the driver's cabin and the temporary affecting of the driver's hearing. Diesel-electric locomotives were introduced on the Rumanian Railroad too recently for us to find out to what extent this auditory fatigue does not lead to occupational deafness with increasing professional exposure.

Solutions That Are Called For

Steps to reduce the noise and vibration level in the driver's cabin of modern locomotives must be addressed to noise sources as well as to cabin design.

Owing to size considerations, the modern engines of diesel locomotives cannot be completely enclosed by a soundproof housing [5]. However, partial acoustic insulation of the upper and lateral surfaces of the Diesel engine is possible with the aid of fixed covers and panels.

For noise reduction in exhaust manifolds different types of attenuators can be employed. /18

Delayed opening of admission valves and delayed closing of exhaust valves, as well as increasing the cross-sectional area of discharge valves, reduced the noise in the admission system.

Experiments conducted in England on Diesel engines have shown that noise can be reduced by 10 dB(A) by building casings with thicker walls made of magnesium alloy and by additional equipping with insulating covers.

In order to reduce the noise of auxiliary devices (compressors, air blowers, fans) it is necessary to cover them with individual housings.

Reduced transmission of noise and vibrations to the driver's cabin requires good insulation against the vibrations of the engine and of all auxiliary mechanisms, which does not, however, exclude insulation of the cabin from the machine room and the

undercarriage. In order to reduce the acoustic energy reflected by the walls, as well as to reduce oscillation energy, it is recommended that special sound-absorbing linings be utilized in locomotive cabins. In principle, separation walls between machine room and cabin must be covered with vibration- and sound-absorbing material.

Summary

The present study is a survey of the data obtained from an extensive study aimed at determining the noise and vibration sources in the modern locomotives of the Rumanian Railroad and their influence on engine drivers. The authors attempt to hierarchize noise and vibration sources in terms of importance and to correlate the noise level with the influence of noise on the engine drivers' organ of hearing. Finally, the authors sketch some possible recommendations for reducing the level of these noxae in order to improve the acoustic comfort of engine drivers. /18

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THE COANDA EFFECT IN GAS-DYNAMIC NOISE CONTROL

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The noxious effects of environmental sound pollution are well- /18
known and the fight against pollution has become one of man's pressing preoccupations.

Finding the most effective means of environmental noise depollution has constituted and still constitutes one of the major preoccupations of the Institute for Scientific and Technical Creation (ISTC). A prominent position in this sphere of preoccupations is occupied by attenuation of the gas-dynamic noise due to the free expansion of gases in the atmosphere, noises that are largely responsible for noise pollution.

In the present state of the art the following categories of silencers are known for gas-dynamic noises due to expansion:

-- active silencers;

-- reactive silencers;

-- multisectioned silencers.

In examining these types, which are well-known from the specialty literature, I shall only mention some of their disadvantages:

-- they attenuate especially high- and intermediate-frequency noises, leaving low-frequency noises practically unattenuated;

-- in order to effect substantial attenuations (on the order of 30-40 dB) they must be large, which sometimes makes them unuseable and greatly increases their cost price;

-- the temperature of the gas jet that can enter the silencer is limited by the characteristics of the sound-absorbing material due to its direct contact with the jet.

In what follows I shall discuss some ISTC accomplishments /19
in this domain, accomplishments that, at present, rank among the most modern and effective means of controlling noise pollution due to gas-dynamic noises of free explosions of steam and gases and that have been patented by ISTC.

All these new types of silencers have common characteristics: throttling the expanded jet, changing its direction (deflection

due to the Coanda effect), bringing the throttled jet close to a Coanda profile, changing the expansion noise frequency spectrum to within the limits for which the silencer's active structure causes maximum attenuation.

From the point of view of how the throttled jet is deflected by the Coanda effect, ISTC silencers can be divided into:

-- silencers with press-down shutters of the Coanda-Teodorescu type;

-- silencers of the Silens-Coanda type with Coanda ejectors.

On the assumption that silencers with Coanda-Teodorescu press-down shutters are well-known, I shall only mention the most representative of that series, the AZ-TM-269 noise silencer for aircraft turbojet engine research stands with a thrust up to 16,000 dN (patent No. 42718/61) invented by a group headed by Professor Constantin Teodorescu. Attenuation of the acoustic pressure level of noise in the 31.5 - 31,500 Hz frequency band ranges from 31 dB to 67 dB.

In what follows I shall discuss the SILENS-COANDA silencers; in the present case, the SILENS-COANDA CET silencer.

The silencer (Fig. 1) consists of a Coanda nozzle of the external type 1, a sound-absorbing body made-up of sections 2 and 3 and evacuation diffuser 4, a silencing screen 5 and a support 6 for the device.

The Coanda nozzle of the external type 1 consists of the support 7, the body 8 equipped on the outside with the Coanda profile 9 and the cone 10. Cut in the body 8 is the ring-shaped cavity a extended toward the exterior by the slit f whose width can be altered by changing the relative position between parts 7 and 8.

For delivery of the fluid that is to be discharged into the atmosphere the device is equipped with a conduit 11, treated for sound absorption for the same length as section 2 of the sound-absorbing body; together with the Coanda nozzle of the external type 1, the conduit forms the central part of the silencer. /19

The SILENS COANDA CET silencer operates as follows:

The gases that are to be discharged into the atmosphere are directed by means of the conduit 11 to the ring-shaped cavity a of the Coanda nozzle 1 and thence to a circular slit f, whence it issues in the form of a fine radial jet.

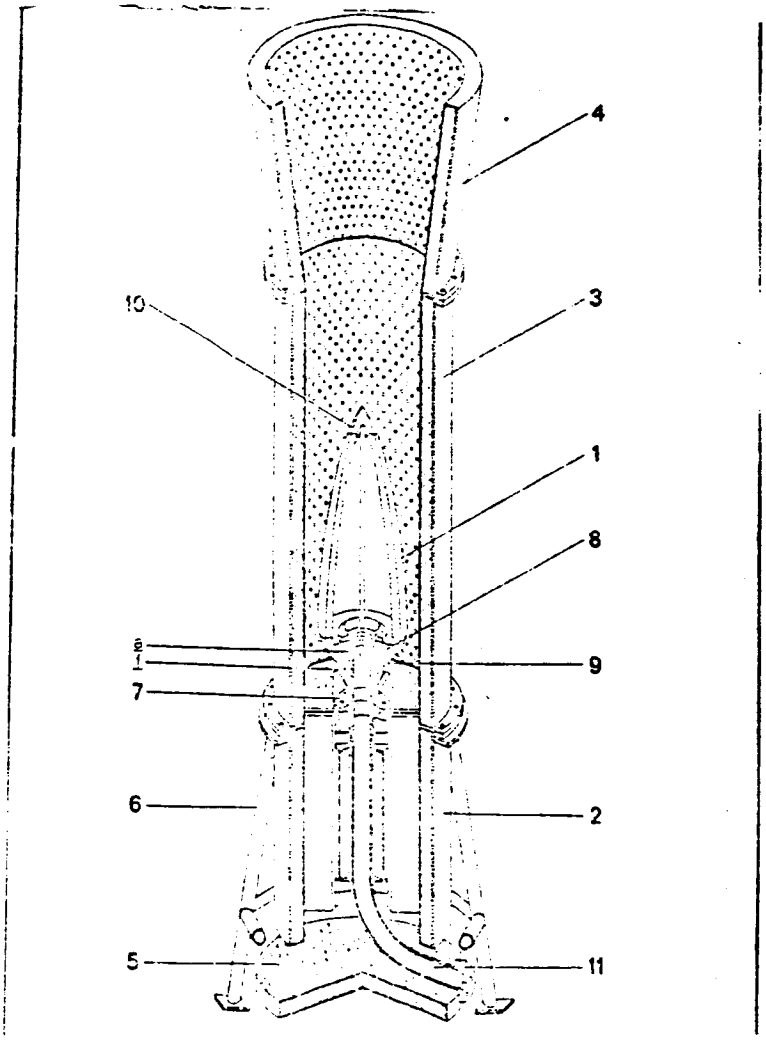

Fig. 1.

Owing to the Coanda effect, the radial jet adheres to the walls 9 of the Coanda jet of the external type, deviating from its initial direction by 90^{o} so as to set up a violent suction of the upstream air.

Because of mixing with the air sucked in from the environment, the hot gases are intensely cooled.

As the gas-air mixture continues to evolve between the walls of section 3 and cone 10, it is continuously braked because of the gradual enlargening of the transversal sections. This braking is continued in the diffuser 4 for the same reason.

The evolution of the gases in the ring-shaped slit f causes a structural change in the noise generated by the jet by displacing the acoustic spectrum to the high- and very high-frequency region, changing, at the same time, the direction of this noise by orienting its predominant components toward soundproofing by sections 2 and 3.

The jet noise with its frequencies thusly altered is readily attenuated by the acoustic structure of sections 3 and 4. In this manner, the acoustic energy of the incident waves is rapidly dissipated and the noise that is propagated downstream of the device is very much diminished.

Similarly, owing to the intense depression existing upstream of the Coanda nozzle, the acoustic waves generated by the jet are considerably impaired in their propagation in this direction.

What is new about this system is that by means of the Coanda effect the fine radial jet sheet is deflected, but not the predominant component of the noise of expansion from the slit, /19
which remains directed toward soundproofing by the active-structure silencer and which has altered frequencies corresponding to the critical flow of the fluid from the slit, and this, precisely in the region where soundproofing by the silencer (2 and 3) shows maximum silencing.

With respect to the flow spectrum in a longitudinal section, the angle between the tangent to the jet's current line and the direction of the predominant component of the expansion noise widens continuously up to 90° downstream from the slit's plane.

Fluid decompression in a Coanda ejector of the external type leads to a shift in direction of the expanded jet and of the predominant component of the noise spectrum, as well as a change in the latter's frequency spectrum because of an increase in the fluid's velocity in the slit.

Consequently:

-- expansion noise can be silenced with maximum efficiency by an active-structure silencer;

-- gas-dynamic noise due to the flow of the turbulent mixture of expanded gas and sucked-in air is diminished by the continuous enlargening of the downstream sections;

-- the temperature of the expanded fluid (in the case of hot gases) is reduced by means of intensive mixing with the air absorbed by the depression created by the Coanda effect.

Acoustic Performance

The results of measurements of the noise level of a superheated steam expansion in a SILENS-COANDA-CET 20/40 silencer and without a silencer are shown in Table 1 and the audiospectrogram of Fig. 2.

TABLE 1: ACOUSTIC PERFORMANCE OF SILENS-COANDA-CET 20/40 SILENCER

Frequency (Hz)	31,5	63	125	250	500	1000	2000	4000	8000	16000	31500	Liniar
Noise level of free expansion without silencer in the CR-12 boiler for 40 ata and 450° C steam for a distance of 25 m (dB).	95	104	103	110	114	120	126	125	123	113	105	132
Noise level of free expansion in SILENS-COANDA 20/40 silencer for 40 ata steam for a distance of 25 m (dB).	80	84	86	78	74	73	74	77	79	78	60	91
Admissible noise level prescribed by the "Republican Labor Protection Norms" (dB).	110	103	96	91	88	85	83	81	80	79	87	-
Attenuation with "SILENS-COANDA-20/40 silencer.	15	20	17	32	40	47	52	48	44	35	45	41

It can be seen that the general noise of the steam jet is attenuated to below the level prescribed by the labor protection norms (Cz85 curve).

Maximum attenuation is 52 dB for 2,000 Hz and mean attenuation is 35-40 dB in a wide-frequency band.

Continuous operation of the silencer incident to starting and stopping means that the jet must follow the ejector's Coanda profile in the pressure range from zero to the fluid's nominal pressure in the slit (the critical pressure). /19

In order to obtain a stable flow in this wide pressure range we had to resolve some difficult problems in the mechanics of compressible fluids.

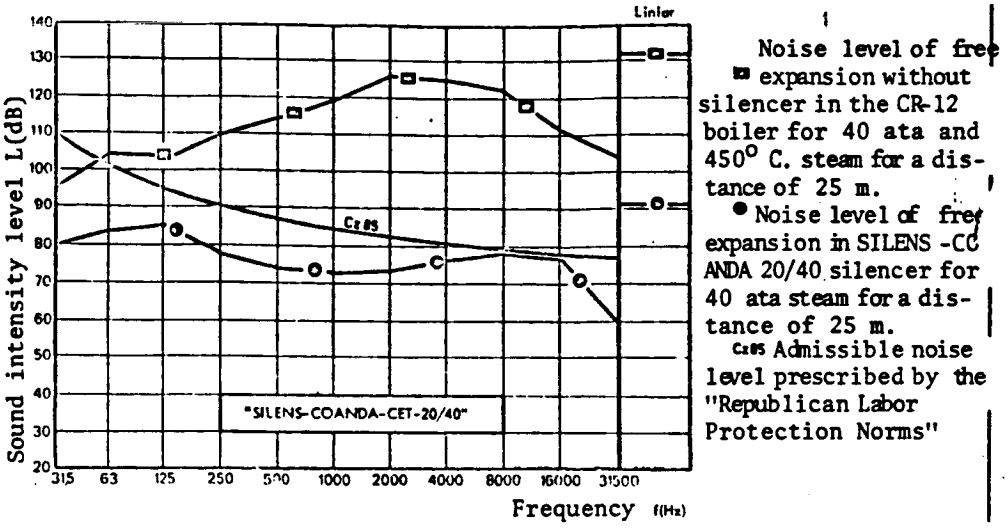

Fig. 2. Acoustic performance of SILENS-COANDA-CET 20/40 silencer

The difficulty is due to the nonlinearity of the basic equations, for which no exact method of solution exists. In order to trace the boundary of the jet in the initial portion (the nozzle outlet section) approximative methods of solution were found, and studies and checks were conducted on experimental models.

Thus, the deflection of the current, of the jet's maximum width and of the wavelength for the first wave was determined. The independent variables are: the pressure at the zero velocity point (tank pressure) and the ratio of the specific heats. The dependent variables are: the Mach number, the angle of velocity with the jet's axis of symmetry and the boundary of the jet.

Summary

The author discusses the principle types of silencers for gas-dynamic noise of free steam and gas expansions, as well as the results of research in the field of the gas dynamics of jets and of applied acoustics conducted by a group from the Institute for Scientific and Technical Creation. /19

Gas-dynamic noise attenuation by means of the Coanda effect is due to fluid decompression in a Coanda ejector of the external type, where a structural change takes place in the acoustic

frequency spectrum and in its direction, as well as a substantial decrease in the fluid's velocity, temperature and concentration. This process is continued in the second phase with absorption of the acoustic waves by means of an active structure.

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CALCULATION METHOD FOR ACTIVE SILENCERS WITH A VARIABLE SECTION

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The noise level in ventilation plants is reduced by using active silencers (cellular or lamellar) with a constant section. The attenuation achieved by these devices is proportional to the perimeter and length of the silencer, the sound absorbent qualities of the material used to manufacture them and inversely proportional to the section. In most cases, especially in the industrial environment, the attenuation that is required is so great in a rather wide frequency band that silencer size becomes appreciable. This fact gives rise to difficulties for designers and builders in placing noise silencers along the conduits of ventilation plants. /19

These difficulties can be partially eliminated, if active noise silencers are built and mounted also along portions of ventilation conduits with a variable section (convergent and divergent sections and elbows). In order to apply this new solution for ventilation plant noise level reduction easily and on a large scale, I have worked out a calculation method for these active silencers with a variable section. This method is the subject of the present communication.

In the case of active noise silencers (generally cellular) with a constant section depending on the length, the attenuation ΔL in dB is calculated with Eq. (1) from reference [2]: /20

$$\Delta L = 1{,}1\,\varphi(\alpha)\frac{P}{S}\ell \quad [dB] \tag{1}$$

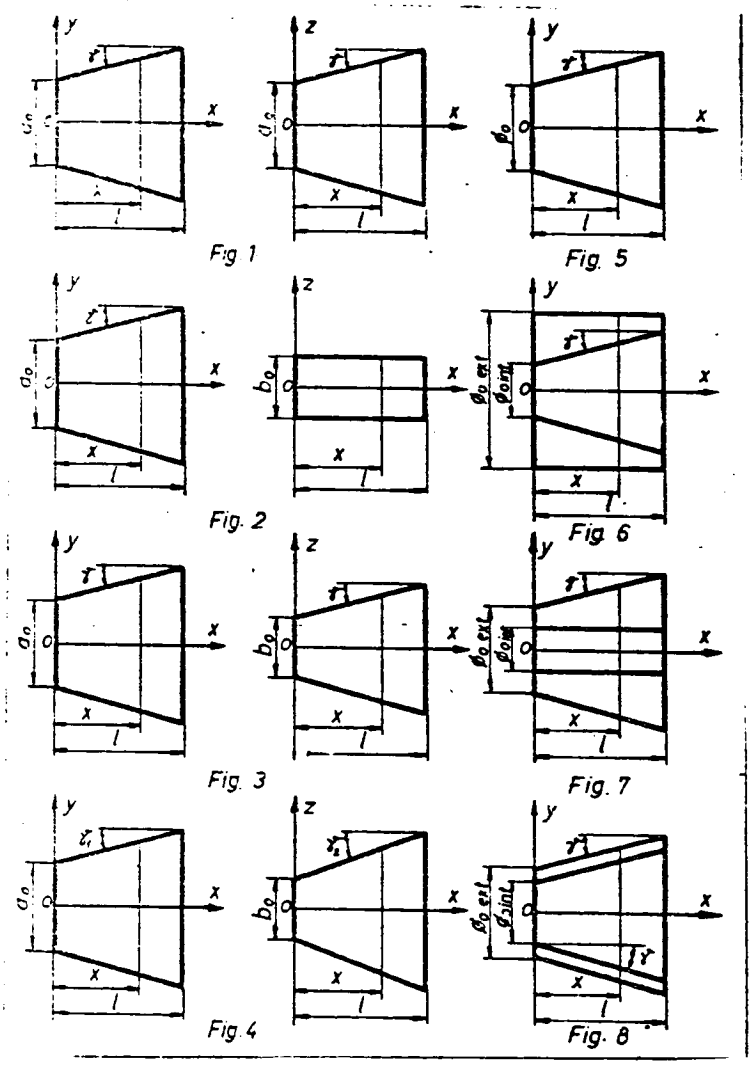
in which: $\phi(\alpha)$ is a function that depends on the coefficient of acoustic absorption of the sound absorbing material used in manufacture; P is the silencer perimeter in m; S is the silencer section in m^2; and ℓ is the silencer length in m.

In the case of ordinary silencers the magnitude P as well as the magnitude S are constant for the silencer's entire length ℓ.

In the event that the silencers have a variable section depending on the length ℓ (frustum of pyramid and of cone), Eq. (1) assumes different forms depending on the silencer's geometrical configuration.

New calculation relations can be derived on the basis of Eq. (1) by means of the following method:

-- a system of rectangular axes Ox, Oy, Oz is chosen with their origin placed at the geometrical center of the small base of the silencer shaped like the frustum of a pyramid or of a cone (Figs. 1-8);



-- knowing that, depending on the manner of choosing the coordinate axes in a plane that is perpendicular to the coordinate axis and that is situated at the distance x from the origin of the axes, both the perimeter P and the section F of the silencer are functions of x, we denote these magnitudes by P(x) and S(x);

-- under these circumstances, for an infinitesimal variation dx of the length x of the silencer, the variation of the attenuation d(ΔL) has the form: /20

$$d(\Delta L) = 1.1\,\varphi(\alpha)\frac{P(x)}{S(x)}dx \qquad (2)$$

-- for the silencer's entire length ℓ between x = 0 and x = 1 the total attenuation ΔL is equal to:

$$\Delta L = \int_0^{\ell} d(\Delta L) = \int_0^{\ell} 1.1\,\varphi(\alpha)\frac{P(x)}{S(x)}dx = 1.1\,\varphi(\alpha)\int_0^{\ell}\frac{P(x)}{S(x)}dx \qquad (3)$$

With the aid of Eq. (3) we can calculate the attenuation produced by any silencer whose shape is represented by one of the Figures 1-8 by calculating for each case separately the ratio P(x)/S(x), which varies with the silencer's geometrical shape.

Using this method, we get the following results:

a. Frustum of pyramid with a square base (Fig. 1):

$$\frac{P(x)}{S(x)} = \frac{4(a_0 + 2x\operatorname{tg}\gamma)}{a_0^2 + 2a_0 x\operatorname{tg}\gamma}\,; \qquad \Delta L = \frac{2.2}{\operatorname{tg}\gamma}\ln\left[1 + 2\frac{\ell}{a_0}\operatorname{tg}\gamma\right]\varphi(\alpha)$$

b. Frustum of pyramid with a rectangular base and two parallel lateral faces (Fig. 2):

$$\frac{P(x)}{S(x)} = \frac{2(a_0 + b_0 + 2x\operatorname{tg}\gamma)}{a_0 b_0 + 2(a_0 + b_0)x\operatorname{tg}\gamma}\,; \qquad \Delta L = \frac{1.1}{\operatorname{tg}\gamma}\left[\ln\left(1 + 2\frac{\ell}{a_0}\operatorname{tg}\gamma\right) + 2\frac{\ell}{b_0}\right]$$

c. Frustum of pyramid with a rectangular base and nonparallel lateral faces but equally inclined at both bases (Fig. 3):

$$\frac{P(x)}{S(x)} = \frac{2(a_0 + b_0 + 4x\operatorname{tg}\gamma)}{a_0 b_0 + 2x\operatorname{tg}\gamma + 4x^2\operatorname{tg}^2\gamma}\,; \qquad \Delta L = \frac{1.1}{\operatorname{tg}\gamma}\left\{\ln\left[\frac{a_0 b_0 + 2(a_0 + b_0)\ell\operatorname{tg}\gamma + 4\ell^2\operatorname{tg}^2\gamma}{a_0 b_0}\right]\right.$$

d. Frustum of pyramid with a rectangular base and nonparallel lateral faces (Fig. 4):

$$\frac{P(x)}{S(x)} = 2\left[\frac{a_0 + b_0 + 2x(\operatorname{tg}\gamma_1 + \operatorname{tg}\gamma_2)}{a_0 b_0 + 2x(b_0\operatorname{tg}\gamma_1 + a_0\operatorname{tg}\gamma_2) + 4x^2\operatorname{tg}\gamma_1\operatorname{tg}\gamma_2}\right]\,;$$

$$\Delta L = \frac{0.55}{\operatorname{tg}\gamma_1\operatorname{tg}\gamma_2}\left\{(\operatorname{tg}\gamma_1 - \operatorname{tg}\gamma_2)\ln\left[\frac{b_0(a_0 + 2\ell\operatorname{tg}\gamma_1)}{a_0(b_0 + 2\ell\operatorname{tg}\gamma_2)}\right] + \right.$$

$$\left. + (\operatorname{tg}\gamma_1 + \operatorname{tg}\gamma_2)\ln\left[\frac{a_0 b_0 + 2\ell(b_0\operatorname{tg}\gamma_1 + a_0\operatorname{tg}\gamma_2) + 4\ell^2\operatorname{tg}\gamma_1\operatorname{tg}\gamma_2}{a_0 b_0}\right]\right\} \varphi(\alpha)$$

e. Frustum of cone (Fig. 5): /20

$$\frac{P(x)}{S(x)} = \frac{4}{\phi_0 + 2x\,tg\,\gamma} \; ; \quad \Delta L = \frac{22}{tg\,\gamma} \ln\left[1 + 2\frac{l}{\phi_0}tg\right]$$

f. Cylinder with a core in the shape of a truncated cone (Fig. 6):

$$\frac{P(x)}{S(x)} = \frac{2}{\phi_{0\,ext} - \phi_{0\,int} - 2x\,tg\,\gamma} \; ; \; \Delta L = \frac{11}{tg\,\gamma} \ln\left[\frac{\phi_{0\,ext} - \phi_{0\,int}}{\phi_{0\,ext} - \phi_{0\,int}}\right.$$

g. Frustum of cone with a cylindrical core (Fig. 7):

$$\frac{P(x)}{S(x)} = \frac{2}{\phi_{0\,ext} - \phi_{0\,int} + 2x\,tg\,\gamma} \; .$$

noting that regardless of the values of $\phi_{0\,ext}$, $\phi_{0\,int}$ and x (with x ranging from 0 to 1), as can also be seen from Fig. 7, we always get:

$$(\phi_{0\,ext} - \phi_{0\,int}) > 2x\,tg\,\gamma \; ; \; \Delta L = \frac{11}{tg\,\gamma} \ln \frac{\phi_{0\,ext} - \phi_{0\,int} + 2l\,tg\,\gamma}{\phi_{0\,ext} - \phi_{0\,int}} \; ;$$

h. Frustum of cone with a cylindrical core (Fig. 8):

$$\frac{P(x)}{S(x)} = \frac{\pi_2\left[\phi_{0\,ext} + \phi_{0\,int} + 4x\,tg\,\gamma\right]}{\pi_4\left[\phi_{0\,ext}^2 - \phi_{0\,int}^2 + 4x(\phi_{0\,ext} - \phi_{0\,int})\,tg\,\gamma\right]} = \frac{2}{\phi_{0\,ext} - \phi_{0\,int}} \; ;$$

$$\Delta L = 2.2\,\frac{l}{\phi_{0\,ext} - \phi_{0\,int}}\,\varphi(\alpha)$$

Testing these equations on a number of experimental models showed good agreement between calculations and measurement results, the maximum differences being on the order of ±2.5 dB.

Summary

The most wide-spread method of noise reduction in ventilation plants uses active silencers with a constant section. In most cases, so great is the required attenuation that the silencers must be of a very large size. This situation causes difficulties in placing them on ventilation conduits. /20

By introducing variable section silencers in divergent and convergent sections or elbows of ventilation plants, the length of classical silencers can be reduced.

The paper presents an original calculation method for active noise silencers with a continuously variable section, as well as calculation relations worked out by this method for eight common practical cases.

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REDUCTION OF DIESEL ENGINE EXHAUST NOISE IN THE PETROLEUM MINING INDUSTRY

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1. Introduction

The crude oil and gas mining industry has undergone rapid development in all oil-producing countries including Rumania owing to the ever-growing demand for petroleum products and the rapid development of the petrochemical industry. /20

This development has necessitated the introduction of a perfected technology, new methods and ever more modern plants, which have contributed to the continuous increase in drilling noise.

During drilling the rotary table and the drill pipe string are driven -- through the agency of different types of couplers -- by two, three, or even four Diesel engines. These engines constitute enginehouse noise sources.

Outside the enginehouse an important noise source is Diesel engine exhaust, which can attain maximum values of 116 dB.

2. Present State of the Art of Diesel Engine Exhaust Noise Control Abroad and at Home

In order to attenuate Diesel engine exhaust noise, which is an aerodynamic noise situated in the low- and high-frequency band up to 1,000 Hz, different types of silencers or mufflers have been proposed.

Thus, for low-power Diesel engines, especially those of tractors and automobiles, silencers can be applied whose damping effect is achieved by means of a well-dimensioned space. /20

Fig. 1 shows another example of a silencer for Diesel engine exhaust.

The silencer consists of a series of differently sized expansion chambers. The connection between the chambers is executed with care so as to impart to it the necessary acoustic properties and not to set up additional resistance to the gas flow.

Fig. 1. Silencer for stationary motors.

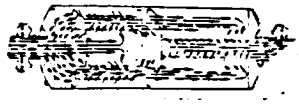

Fig. 2. Burgess silencer.

In the event that the Diesel engines serve a drilling installation where the problem of protecting it against fire may arise the noise silencers can be equipped with spark catchers(Figs. 3 and 4).

The Vortex silencer is so-designed that the gases that enter it are guided by guide vanes that are constructed so as to form convergent orifices that impart a helicoidal motion to the gases. The transported gas particles are projected toward the walls of the Vortex chamber and are ejected through discharge orifices into an air space in the dust channel, where they drop by gravity. /20

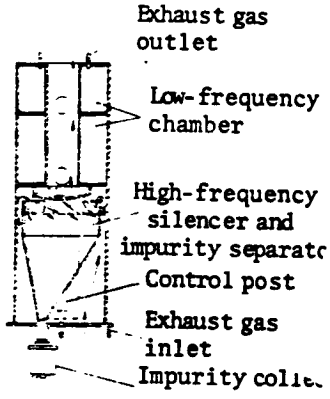

Fig. 3. Maxim double silencer.

Fig. 4. Vortex silencer with spark catcher.

3. Design and Calculation of a Silencer Applied to a Diesel Engine in a Rumanian Drilling Installation

As a result of noise determinations conducted at the exhaust holes of Diesel engines in drilling installations, it was found that the noise level exceeds the admissible Cz85 limiting curve by maximum values of 20 dB.

In order to reduce the exhaust noise level, a reactive silencer of the derivative resonator type was designed. In reactive silencers sound absorption is assured by formation of a "wave stopper" that impedes the passage of the sound at some frequencies because of the influence of the mass and the elasticity of the air in the silencer's cells. The silencer is based

on the principle of acoustic filters. Characteristic of these systems is their capacity to let pass without appreciable damping those oscillations that have frequencies in one or more ranges and annul or reflect toward the source all those oscillations that have frequencies outside of these ranges. From the acoustic point of view, this solution consists of a central conduit through which the exhaust gases pass and a series of cavities with rigid walls placed around the central conduit in a concentric manner. Each cavity communicates with the central conduit by means of a connecting orifice (Fig. 5). These cavities act like decompression chambers for the gas in the conduit's section.

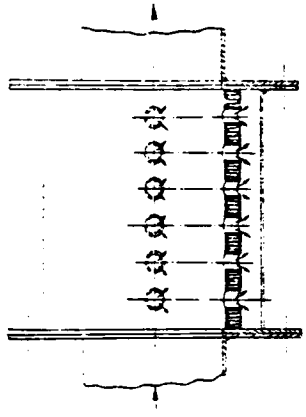

Fig. 5. Reactive silencer applied to Diesel engine exhaust in a drilling installation.

Calculation Method

The silencer's real attenuation ΔL_r in dB is calculated as follows: /208

1. The silencer's resonance frequency f_0 in Hz is calculated. Chosen for the resonance frequency is the value corresponding to the necessary attenuation δL, of maximum value, which is checked by calculating the value of the real attenuation ΔL_r versus the frequency (see points 2 and 3). If the resonance frequency is correctly chosen the calculated value of ΔL_r versus the frequency must be greater than or equal to the real attenuation δL in the whole frequency band in which $\delta L > 0$. Should this condition not be satisfied, we can choose for the resonance frequency a value equal to that of the second or third harmonic or of one of the subharmonics of the frequency corresponding to the attenuation δL of maximum value.

2. Knowing the resonance frequency f_0, with the aid of relation:

$$\frac{\sqrt{K_1 V}}{2S_1} = \sqrt{\left[10^{0.1\,\delta L_{inf}} - 1\right] \cdot \left[\frac{f_{inf}}{f_0} - \frac{f_0}{f_{inf}}\right]^2} \qquad (1)$$

the value of the ratio $\frac{\sqrt{K_1 V}}{2S_1}$ is determined. The different parameters of this ratio have the following meanings:

K_1 is the admittance of the connecting orifice between the silencer's central conduit and the afferent resonant cavity in cm;

V is the total volume of the resonant cavities in cm^3;

S_1 is the central conduit's section in cm^2;

δL_{inf} is the necessary attenuation corresponding to the lower frequency in dB. The value of this attenuation is equal to the minimum value of the attenuation δL for which $\delta L > 0$;

f_{inf} is the frequency corresponding to the attenuation δL_{min} in Hz;

f_0 is the resonance frequency in Hz.

3. With the aid of the relation:

$$\Delta L = \left[\frac{\sqrt{\frac{K_1 V}{2S_1}}}{\frac{f_{sup}}{f_0} - \frac{f_0}{f_{sup}}} \right]^2 \qquad (2)$$

the real attenuation at the higher frequency f_{sup} is calculated.

4. With the aid of the relations: /20

$$K_1 = \frac{nS_0}{\ell_0 + C_1 \sqrt{S_0}} \quad [cm]$$

$$\ell_0 = \sqrt{\frac{4S_0}{\pi}}$$

$$\frac{K_1 V}{2S_1} = \frac{\pi \phi_{int}^2}{4} \cdot \frac{K_1 V}{2S_1} \quad [cm]$$

$$V = \frac{\pi}{4} \cdot L (\phi_{ext}^2 - \phi_{int}^2) \quad [cm^3] \qquad (3)$$

in which:

a is the number of orifices;

S_0 is the section of an orifice in cm^2;

L is the silencer's length in cm;

ℓ_0 is the length of an orifice in cm;

ϕ_0 is the diameter of an orifice in cm;

ϕ_{int} is the diameter of the silencer's conduit in cm;

ϕ_{ext} is the silencer's outer diameter in cm, the silencer's acoustic parameters are determined. The method of calculating these parameters is by trial and error. In choosing the values of certain parameters we must see to it that the ratio of two parameters will not be equal to a multiple or submultiple of a whole number.

With the aid of these data and of Eq. (2), in which the frequency of f_{sup} is replaced by frequency f and the attenuation ΔL by the real attenuation ΔL_r, the real values of a silencer's attenuation ΔL_r in dB is calculated. The calculated values of the real attenuation attain values up to 26 dB for a frequency of 250 Hz.

Such a calculated silencer was applied to the exhaust of a Diesel engine in a drilling plant. Fig. 6 shows the noise level before and after applying the silencer to the exhaust. The hatched portion represents the attenuated acoustic energy. Unrealized attenuations are largely due to non-observance of the indicated design data. /21

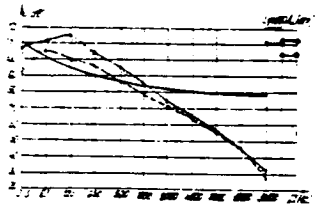

Fig. 6. Attenuation by a reactive silencer installed in a Diesel engine exhaust.

4. Conclusions

From this study of Diesel engine exhaust noise control we can draw the following conclusions:

1. The noise level in a Diesel engine exhaust exceeds the admissible Cz85 limiting curve in the low- and high-frequency region (up to a frequency of 2,000 Hz) by maximum values of 20 dB for a low frequency of 125 Hz.

2. A high noise level has a noxious effect on the human organism, so that it must be reduced at all costs.

3. Correctly choosing and applying the proposed solutions will lead to a considerable drop in the noise level.

Summary

Noise has a noxious effect on the human organism. In order to /21 control it a continuous war is waged.

An important noise source in a drilling plant is Diesel engine exhaust. In order to reduce this noise, a reactive silencer of the derivative resonator type was proposed, calculated from the

acoustic and design point of view and applied. As a result of applying such a silencer on the exhaust conduit of a Diesel engine the noise level dropped down to 18 dB.

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N74-28389

THE PROBLEM OF NOISE IN WEAVING MILLS

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1. Generalities

The increase in the consumption of fabric has inspired a rapid development of weaving mills both in terms of the building of highly perfected looms with increased speeds (picks/min) and by a great concentration of implements, which has led to the construction of large-capacity weaving mills. /21

These advances have been accompanied by such an increase in noise pollution that the total noise level in weaving mills has high values (102-104 dB/A) and the spectrum of this noise is distributed especially in the high-frequency bands, for which reason it has an extremely noxious effect on the human organism. This fact is of particular importance if we consider that a great number of employees, especially women, work in weaving mills.

As regards the effects of noise on production, recent studies have brought to light that, owing to the phenomenon of nervous fatigue, the attention is diminished in production sections with a high noise level and, as a result, working efficiency and the quality of the products drop, the possibility of accidents and fluctuation in personnel increase , etc., which is reflected in the cost price of the products. By some researchers the effect of noise on working efficiency is estimated to be 6-20%.

In view of the importance of this problem, the Institute of Engineering Designing of Light Industry, in collaboration with the Acoustics Commission, INCERC and, especially, the Bucharest Polytechnic Institute in as early as 1965 initiated a series of studies and experiments divided into several stages for the purpose of determining existing noise levels in different sectors of light industry and, especially, in weaving mills, as well as finding the most efficient solutions for noise reduction. /21

2. Noise Measurements and Their Interpretation

As the first stage in determining the noise level existing in weaving mills and in establishing the influence of some factors that depend on the type of construction, the number, type and manner of operation of the weaving looms, etc., noise measurements were conducted in a number of weaving mills.

The results of these measurements were made the object of some studies that brought to light a number of findings [1, 2], namely:

a) In all weaving mill rooms where measurements were conducted, the noise level due to the operation of the weaving looms exceeds the Cz85 curve as well as the "B" discomfort curve.

Spectral analysis has shown that predominant in weaving mills are high-frequency components greater than 1,000 Hz, so that the influence of the noise on the human organism is particularly noxious.

b) The noise level due to the operation of a single shuttle loom placed in the open air exceeds the Cz85 curve.

c) The number of weaving looms has very little influence on the total noise level: for a ten-fold increase in the number of looms the increase in the sound level is 2-3 dB.

d) On the assumption of many weaving looms operating in a closed room, the increase in the sound level is due, in the first place, to reverberation and to a lesser extent to the number of weaving looms.

e) The influence of the type of construction on the total noise level is insignificant. An increase in the sound level of about 2-3 dB appears in storied rooms, which is due to the vibrations transmitted by solid conduction from other stories. /21

3. Solutions for Noise Reduction

As a result of studies conducted in the first stage, it appeared that in weaving mills equipped with shuttle looms, the noise level is very high, and, on the other hand, noise reduction measures applied directly at the source by different machine building enterprises did not yield satisfactory results (reduction being on the order of 2-3 dB), for which reason experiments were conducted with a view to more substantial noise reduction in weaving mills.

In the present study, some solutions for noise reduction will be discussed that were applied in an experimental manner, as well as the results obtained.

a) Noise reduction measurements applied to buildings

In order to reduce the noise level in a number of weaving mills of different countries, sound-absorbing treatment of walls and, especially, the ceiling, has been applied.

With a view to checking the efficiency of this solution, the Institute of Engineering Design of Light Industry in collaboration with the Polytechnic Institute worked out and tested in the period 1965-1970 a number of sound-absorbing structures made of materials that were being produced in Rumania at the time, namely:

-- a sound-absorbing ceiling made of centrifuged mineral wadding cushions covered with perforated PFL plates in the Satu Mare ITI Weaving Mill. The noise attenuation achieved with the aid of this type of ceiling was about 2-5 dB.

-- a sound-absorbing ceiling structure made of cassettes of perforated aluminum and 50 mm sillan plates (Fig. 1) was applied in a number of weaving mills after 1970, when the respective materials began to be manufactured in Rumania.

This type of sound-absorbing platform was applied in an experimental form in the Iasi "Victoria" Weaving Mill with an area of about 10,000 m^2 and initially equipped with 540, and finally 780 weaving looms from Rutti si Unirea, Cluj. /21

Measurements conducted before and after application of the sound-absorbing platform by the Bucharest Polytechnic Institute and INCERC [3, 4, 5] are shown in the spectrogram of Fig. 2.

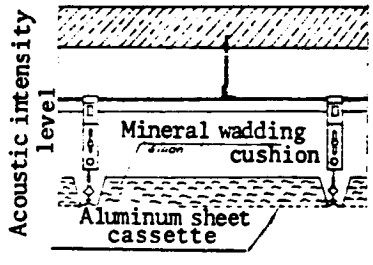

Fig. 1. Sound-absorbing platform applied in the Iasi Victoria Weaving Mill.

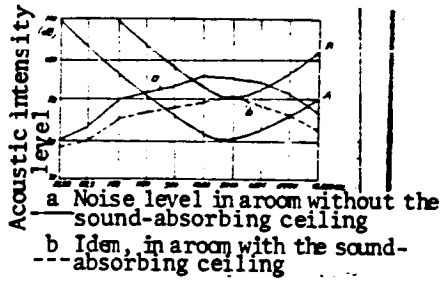
Acoustic intensity level
a Noise level in a room without the sound-absorbing ceiling
b Idem, in a room with the sound-absorbing ceiling

Fig. 2. Noise spectrogram in the Iasi Victoria Weaving Mill before and after application of the sound-absorbing ceiling.

Analysis of this spectrogram shows that noise was reduced by 6-7 dB, so that the noise level was lowered to the B discomfort curve. On the other hand, no decrease in the noise level to the admissible level was obtained because, as was mentioned at Point 2, the level of noise generated by the operation of a single shuttle loom placed in the open air exceeds the Cz85 curve.

Then again, it was found that the reverberation time was lowered to acceptable values (from 3.4 to 1.5 sec).

As regards productivity, it was estimated that by applying this ceiling productivity could be increased by about 12%.

b) Noise reduction measures at the source were applied by replacing classical looms by shuttleless looms [6].

The most efficient solution for reducing noise at its source consists in replacing the actuators of the shuttle looms with shuttleless looms that run much more silently.

Taking into account both this fact and the technological process, provision was made for equipping some weaving mills with compressed-air looms and water-jet looms. /21

In the spectrogram of Fig. 3 the values of the noise levels existing in a weaving mill room equipped with the same number of classical looms are compared with those for compressed-air looms.

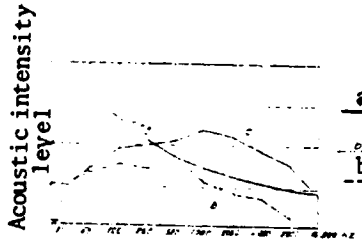

Fig. 3. Spectrogram of noise produced by operation of ATPR looms as compared with that of weaving looms.

c) Elastic emplacement of weaving looms by means of vibration damping devices, especially in the rooms of storied weaving mills, contributes to noise reduction. By mounting some weaving looms on special vibration shock-absorbers an attenuation of 2-3 dB was obtained [7].

d) Individual protection measures

In the rooms of existing weaving mills, where noise reduction measures cannot be applied, MIU has initiated the experimental application of individual protection measures and, in particular, soundproof earplugs have reduced the noise by about 8-12 dB.

Conclusions

a) The shuttle looms now in use in weaving mills produce a noise level that exceeds the admissible level represented by the Cz85 curve as well as the B discomfort curve.

b) The level of noise generated by the operation of a single shuttle loom exceeds the Cz85 curve. /21

c) By applying a sound-absorbing ceiling in weaving mills equipped with shuttle looms it is not possible to reduce the noise level to the Cz85 curve, but an approximately 60% reduction in the intensity level in sones is obtained, as well as a drop in the reverberation time down to acceptable values, which leads to a considerable improvement in the working conditions in weaving mills.

d) Further research is required with a view to bringing about a more substantial noise reduction right at the source, so that by combining this measure with sound-absorbing treatment of buildings, the noise level in weaving mills can be lowered to the neighborhood of the Cz85 curve.

e) In the design stage, when the implements with which the new weaving mill sections are going to be equipped are determined, the possibility of equipping these weaving mills with shuttleless looms must also be analyzed from the technological and economical points of view.

In this connection, it is necessary that by continued research of our own or by borrowing, shuttleless looms be developed in Rumania.

f) Employees who work in existing weaving mills should be provided with antinoise earplugs and at the same time efforts should be made to convince them of the need to employ individual protection measures.

g) The application of noise reduction measures in weaving mills is fully justified by the substantial improvement in working conditions that entail an increase in productivity and a decrease in additional expenses due to the existence of a high noise level, so that the sums required for application of noise reduction measures will be amortized in a very short time.

Summary

In weaving mill departments equipped with shuttle looms there exists a high noise level that has a noxious effect on the human organism, a fact which affects a very great number of employees and, especially, women and at the same time has negative repercussions on productivity. /22(

Taking into account the importance of this problem, the IPIU in collaboration with the Acoustics Commission, INCERC and, especially, the Bucharest Polytechnic Institute in as early as 1965 initiated research and experiments that were conducted in two stages:

-- Measurements were conducted in different weaving mills to establish the noise level as well as the influence of some factors that depend on the type of construction, the type, number and manner of operation of the looms, etc.

Solutions for noise reduction were established that were applied experimentally in some weaving mills, and the results obtained are discussed in the present study.

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LEVEL, PECULIARITIES AND EFFECTS OF COAL MINE NOISE ON PIT WORKERS

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I. In spite of the prevelance of noise in modern life and in industry and in spite of all the attention that has already been devoted to controlling it, noise remains a major problem that has not yet been solved from many points of view. Specialists are confronted with the following difficulties [1]: /22

-- the clinical and experimental study of its influence on the organism has not been concluded. Occupational deafness still conceals many unknowns: individual differences, its relationship with presbycusis, etc. Noise-induced disorders that are unspecific, obvious though they are, get lost in the larger picture of our general wear, a very complex problem;

-- protection against noise requires heavy materials and costly studies; at first blush one might even say that the success of soundproofing is conditioned by the volume of building materials. In the case of machines, there exists a parallelism between productivity and noise level;

-- soundproofing in industry and town planning must still be envisaged and applied from the conceptual point of view because otherwise it becomes illusory.

In the last few decades, the world's mining industry has not accorded the same importance to noise and vibration control as it has to dust control.

At the behest of the Minister of Mines, Petroleum and Geology, the Mining Safety Research Center, in collaboration with the Labor Protection Research Center, in a number of studies has investigated the level and characteristics of pit noise and its noxious influence on pit workers [2].

The rapid pace with which the Rumanian mining industry is being expanded and modernized obliges us to intensify our future efforts to know and control this noxa. /22

II. Working Method

In order to attain our objective our investigations were carried out in parallel: in the field in order to evaluate the

physical and physiological characteristics of pit noise; and in the laboratory in order to determine specific reactions of the miners' organisms to this noxa.

The noise level was determined in active work spaces (galleries, coal faces, etc.), as well as in less crowded work spaces (pump room, pit bottom, etc.) of the Valea Jiului and Motru mining basin, with the aid of a Brüel device equipped with an octave filter, by means of which we recorded the intensity, frequency spectrum and duration of noise emission at each work space and for each implement.

The results were compared with the maximum limits set by the norms in force.

The hearing threshold was determined with the aid of an EM40 ATLAS audiometer, and in interpreting the audiogram we made corrections for age, neglecting to process cases with ear affections or head injuries with repercussions on hearing.

III. Experimental Results

3.1. Level and Peculiarities of Pit Noise

The sound environment in the pit is the result of the noise of all machines and implements that are being operated.

Each noise source has its own frequency spectrum, by means of which it can generally be distinguished from the others. An analysis of these spectra is given in Table 1.

The noise level in the pit is not uniform; it is continuously modified as the function of mining methods, degree of mechanization, machine operating conditions and the coal bed that is being mined. Simultaneous operation of all the machines in a given work space /223 leads to very intense noises, with compound frequency spectra. For example, in a mechanized working the following can be operating simultaneously: the cutter-loader, the conveyor belt, the ventilator; in a chamber working: air hammers, the ventilator, the conveyor belt; etc.

All these situations stamp mine noise with certain peculiarities that must be taken into account when setting up technological and medical measures of control.

Owing to the relatively small size of the work spaces in the pit (workings or galleries) noises are powerfully reflected from the walls, so that they are uniformly dispersed throughout the section of the work spaces, and the miner is caught in this diffuse transmission field of the final noise. In narrow work spaces (under 1 m) this phenomenon is more obvious.

TABLE 1: RESULTS OF NOISE MEASUREMENTS

Exp. No.	Implement	Total acoustic pressure level (Lg, dB)	Noise intensity level (phones)	Frequency of noise components with maximum acoustic pressure levels (fmax, Hz)	Pressure levels corresponding to fmax (dB)	Frequencies of components whose levels exceed the admissible C85 limits (Hz)
1.	CP-24 Drill hammer	118	117	250; 500; 4.000; 8000; 2000; 1000	119; 114; 105; 104; 106; 106	125-16.000
2.	CA-15 Pick hammer	101	100	500; 4000; 6000; 8000; 1000.	95; 85; 84; 84; 88.	500-16000
3.	Ø400 Pneumatic ventilator	97	95	1000; 8000; 500	90; 84; 90	500-8000 500-1000
4.	Ø700 Electric ventilator	97	96	500; 1000	94; 90	250-8000
5.	WPD-1 Drilling machine	115	114	2000; 1000; 4000; 500; 8000; 250	108; 108; 101; 106; 94; 95	
6.	PK-3 Cutter-loader	98	89	2000; 1000; 500	89; 89; 90	500-2000
7.	MIP-2 Loading machine	114	107	2000; 1000; 500; 4000; 250	95; 96; 98; 90; 96	250-8000

The power of reflection is, in turn, conditioned by the constitution and structure of the rocks that are being worked. Coal absorbs and attenuates a great quantity of the noise generated, in contrast with headings in dross or ore mines, where the hardness of the rocks causes the noise level to increase as a result of more powerful reflection phenomena.

Depending on the manner of emission, pit noise can have an intermittent character, a pulse character (air hammers) or a continuous character (ventilators, cutter-loaders, etc.).

The miner's skill can influence the duration of exposure to noise: a good miner tries to obtain the maximum of ore with few hammer blows in a short time.

Most exposed to the noxious action of pit noise are the following professions: drillers, miners on the group level, pump and compressor mechanics, signal men, etc.

Audiograms for these professions show that the level, peculiarities and duration of noise varies with the work space in the pit and in the same work space from one moment to the next.

3.2. Effects of Noise on Pit Workers

Certainly, it is a fact that noise has a negative effect /224
on all functions of the organism and, therefore, on the worker's productivity, yet very few data are available concerning its non-specific effects.

On the other hand, audiometric studies have brought to light considerable hearing losses. In 1965 Lumio found that 21% of 1,802 pit workers showed hearing losses greater than 50 dB at frequencies ranging from 500-4,000 Hz. Similar findings have been recorded by Filin, Mas [3], Jirak [4], Hazewood [5], etc.

Our research on the specific effects of noise show the following:

Fifteen percent of the miners from Valea Jiului show auditory fatigue at the end of their shift, with noisy operations occurring in the first part of the shift. In lignite mines the phenomenon is encountered more rarely: 5% of the cases.

On the whole, the audiometric curves bring to light two apparently opposed but identical phenomena. In the left half of the audiogram, for the conversational frequency level (500-2,000 Hz) an adaptation can be observed that finds expression in an insignificant rise in the hearing threshold (10-30 dB) which betrays a resistance, an inertia of the organ of hearing with regard to noise.

In opposition to this phenomenon, determinations at high frequencies (2,000-4,000 Hz) show the presence of scotoma that is specific to occupational hypoacusias, which progresses as a function of the intensity and duration of exposure.

If hypoacusia is defined as beginning with a hearing loss greater than 30 dB, then 31.2% of the 708 miners examined have impaired hearing. Together with reduction in the acuity of hearing, i.e., with accentuation of the degree of deafness, the number of invalids decreases, so that only 1.6% of the workers show a hearing loss of 80 dB.

As a function of age, hearing losses greater than 30 dB are /225
encountered in 14.3% of the groups aged 21-25 years and in 36.5% of the groups aged 41-45 years. The more advanced the age at which the worker begins to be exposed to noise, the more rapidly does hypoacusia set in, as compared with young workers.

In dross miners this impairment appears after five years of exposure, while in coal miners it sets in only after 8-10 years.

We could detect no negative effects of noise in miners working in the mechanized complexes of the Motru basin which were introduced five years ago; and the sound-absorbing capacity of lignite is very great.

Fourteen percent of those with other professions in the pit and a length of service up to ten years show signs of hypoacusia, while 26.6% of those exceeding this length of service show signs of auditory trauma. An exception to the rule are pump and compressor mechanics, in 48.8% of whom signs of deafness appear even though the noise level exceeds but little the admissible limits.

The foregoing leads to the conclusion that occupational hypoacusias are frequent in the mining industry.

Only 6.3% of the 31.2% with hypoacusia involve conventional frequencies; these cases, we believe, ought to be pensioned off or transferred to another job because of the risk of mining accidents and the importance that hearing assumes in executing tasks under the conditions of darkness in the pit, where it takes over some of the sensory load of the eye.

Classically hypoacusia begins at frequencies of 4,000 Hz, but often enough (20%) at 2,000 Hz as well, which means that the cochlea's fragile zone extends over a greater range [6]. Subsequently, deafness sets in in the well-known steps of deepening and enlargening of the scotoma [7, 8, 9].

IV. Taking Steps to Reduce the Noxious Effects of Noise in the Mining Industry

Hearing protection is inspired not only by humanitarian motives, /2
but also for reasons of safety and working efficiency. Countries with advanced mining have plans comprising concrete objectives of noise reduction on the technological, engineering and medical fronts, aiming at different stages of design, construction and running, as well as from the organizational point of view. In this connection, we propose the following:

-- creation of a departmental commission on noise control in the mining industry within the framework of preoccupations with labor protection and prevention of occupational diseases;

-- determination of the sound level and pegging out of the work spaces where this exceeds the maximum admissible limits;

-- application of noise-reducing solutions in situations where the noise level exceeds the maximum admissible limits;

-- introduction of individual means of protection in those work spaces where it was not possible to reduce on the technological front the noise level down to the admissible limits;

-- introduction of audiometric tests incident to the prehiring medical examination and of periodic preventive medical check-ups;

-- intensification of noise control education and propaganda by means of lectures, posters, films, etc.

Summary

Complex is the nature of pit noise, which varies from one work space to another and from one operation to another. The mining method, the degree of mechanization, the composition and structure of the deposit, as well as the section of the work space in which the activity is carried out, impart to pit noise certain peculiarities. /229

There exists a correlation between noise intensity and spectrum and degree of hearing impairment in the different professional categories in the pit. Most affected are dross miners and drillers. In these, the first degree of hypoacusia sets in within the first five years of exposure.

Deafness begins at a frequency of 4,000 Hz, but often enough (20%) at 2,000 Hz as well, progressively evolving through the classical stages if exposure to noise continues.

In the coal-bearing basins investigated, the noise level is, on the whole, 31.2%, but the conventional zone is not involved by more than 6.3%.

Technological and medical measures are proposed for the purpose of avoiding these troubles in the future.

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NOISE CONTROL OF PNEUMATIC PERCUSSION DRILLS

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1. Introduction

Owing to the special working conditions in the pit, for the mining industry perhaps more than for any other branch of industry, the noise generated by implements and machines constitutes a pronounced and many-sided danger. /231

The specific noise situation in the pit depends more particularly on the nature of the work spaces, which in part have surfaces with powerful sound reflectivity.

Owing to the relatively small sections of the galleries and workings, as well as the fact that the lateral surfaces ensure reduced sound absorption, we must reckon with very great participation of reflected waves in the total noise generated at any work space. For this reason, for the same machine and the same operating conditions in the pit, the noise level can increase to values of 6-8 dB.

In the pit we do not, as a rule, encounter continuous generation of noises, which evolve in time very rapidly, depending on the machine and work space.

It has, likewise, been found that by its constitution and structure, the mined material influences the noise level at the work space. Thus, for example, broken-up coal or ore has a relatively good absorbtivity in comparison with the compact coal or ore that go to make up the walls of mining work spaces. /232

Among principal noise sources in the pit a place of honor goes to the pneumatic percussion drill, whose noise level attains values of 117-120 dB. In the present study we propose to analyze the present state of the art of noise and vibration control in the mining industry at home and abroad.

2. Present State of the Art of Noise Control Abroad

At the present time, the principal means of noise control throughout the world are: elimination of noise sources, attenuation of noise at the source and its insulation or absorption.

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The first studies were aimed at attenuation of exhaust noise inasmuch as this is the principal component of the drill's noise.

By using noise silencers of the "pass down" filter type total noise levels have been reduced by 5-6 dB. Such drills are already being manufactured by the following firms: Atlas-Copco (Sweden), Joy (USA), Uzhina Pneumatika (USSR), etc.

In addition to efficient damping of the noise generated by drills, the solutions advanced by the Atlas-Copco firm (Sweden) satisfies other necessary conditions, such as keeping to an optimum operating efficiency, certain weight limits and robustness.

Fig. 1 shows the noise spectrogram recorded for the BBD-91 drill (Panthere, Atlas-Copco, Sweden) without the silencer (curve 1) and with the silencer (curve 2). /233

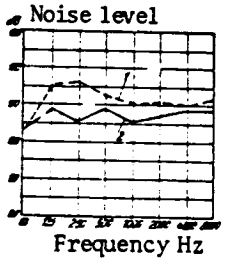

Fig. 1.

Recently introduced on the market in the Soviet Union is the PR24 and LU200 drill equipped with a noise silencer and antivibration device on the drill's handle (Fig. 2), where 1 is the oscillating handle; 2 is the drill body; 3 is the support brace; 4 is connecting rods; 5 is the handle; 6 is the pneumatic support; 7 and 8 are the drill's principal air supply hose; 10 is the noise silencer.

Experiments have shown a reduction in the total noise level by 5 dB.

Preoccupied with the problem of noise control of drills are the following firms: Joy (USA), Tampella (Finland) and Meudon (France).

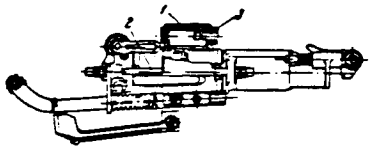

Fig. 2.

3. State of the Art in Rumania

With a view to producing in Rumania drill hammers with a reduced noise and vibration level, research has been conducted in the three following directions:

-- production of a pneumatic motor both simple and silent;

-- production of a small but efficient noise silencer;

-- design of a light vibration silencer. /234

The great majority of drills utilized in the mining industry are equipped either with poppet valve air distribution or slide

valve air distribution. Poppet valve air distribution can be with an oscillating valve (Fig. 3a) or a valve with rectilinear dis-displacement (Fig. 3b).

Fig. 3a.

Fig. 3b.

Among the principal disadvantages of poppet valve distribution are the following: relatively high compressed air consumption; injudicious distribution of the compressed air, the air being evacuated into the atmosphere at a relatively great pressure; reduced mechanical work because of the air cushion that is formed opposite the piston incident to the working stroke; intense noise during operation. The principal disadvantage of slide valve distribution (Fig. 4) is the complexity of the component parts of the distribution mechanism.

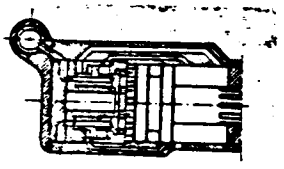

Fig. 4.

With a view to eliminating the disadvantages of both distribution systems, a new kind of free piston motor with self-distribution, recognized as an invention by the OSIM, was calculated, designed and built.

In comparison with poppet valve or slide valve distribution, the self-distribution drill has the following advantages: rational distribution of compressed air by using its potential energy as it is decompressed; inasmuch as the air is evacuated at a smaller pressure, noise during operation is lower; the number of component pieces is smaller. /235

As far as recoil and vibration damping is concerned, all drills equipped with such silencers have them placed in different ways outside of the cylinder, whereby the drill's weight and size are increased.

The P-60 S drill (Fig. 5) with a reduced noise level that we designed, built and tested at the Sibiu "Independence" Factory has the following important characteristics: air distribution is effected by means of the piston's stem; a lower noise level as a result of the presence of the silencer and the fact that the air is evacuated into the atmosphere at a reduced pressure; the recoil silencer is placed inside the cylinder in the form of a mobile cover.

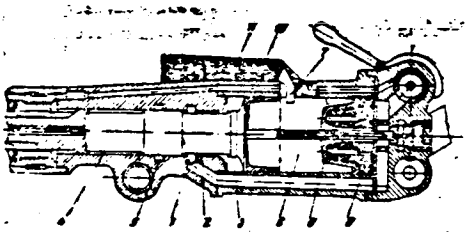

Fig. 5.

The operating principle of this drill is as follows (Fig. 5): at the start of the return stroke, the piston is located at the limiting position on the left. The compressed air from the network penetrates by means of the channel 1 and the ring-shaped space 2 into the cylinder's chamber 3, and the piston is displaced toward the right as it executes its return stroke. At the start of the working stroke, the piston is located at the limiting position on the right. The compressed air penetrates by means of the channel 1, the ring-shaped space 4, the channel 5, then through six longitudinal channels into the cylinder's chamber 6, and the piston is displaced toward the left as it executes the working stroke. /236

Noise measurements conducted during drilling have shown that this type of drill, as compared with other types, has a noise level that is lower by 4-11 dB in components and by 7 dB on the total level.

From the point of view of weight, size and power losses the P-80 S drill, with its noise damping due to the presence of noise silencers, can be compared with the world's most successful models.

Summary

The authors of the present study describe the noise sources in pneumatic drills, bringing to light the fact that air exhaust is the most important source. Then they discuss the present state of the art of noise control of pneumatic percussion drills abroad, indicating the different solutions adopted in this respect. /237

The last part of this study is devoted to the P-58 S and P-60 S drills produced in Rumania and the results of noise measurements.

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MEASURES FOR THE REDUCTION OF THE NOISE AND VIBRATION LEVEL OF APARTMENT HOUSE ELEVATORS

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1. One problem connected with improving the comfort in apartment houses is reduction of their noise level. Often elevators generate noise that exceeds the admissible level, especially in apartments located near the elevator shaft and the winch chamber. /23

Measurements have shown that situations can arise where the noise level in the winch chamber exceeds the Cz65 curve established by the norms in force.

Research has brought to light that the principal noise and vibration sources of the elevator are:

-- the driving winch;

-- the battery of contacts mounted on a panel in the winch chamber;

-- the cabin's mobile control cam.

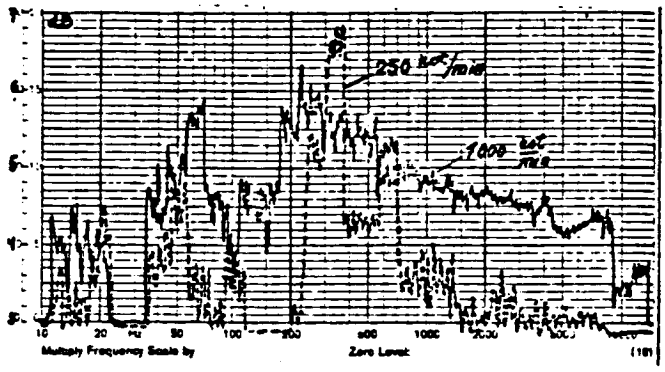
2. Noise sources were analyzed separately in order to estimate the contribution of each one of them to the total noise and vibration level.

As far as the winch is concerned, measurements were conducted at the motor's two operating speeds (250 and 1,000 rpm) as well as for the transitory regimes corresponding to variations in speed.

a. The intensity level of the noise generated by the winch at the motor's two speeds was recorded; the spectrograms are shown in Fig. 1.

b. The transitory regime of change in the motor's speed leads to the appearance of sound pulses with values of about 30 dB. Fig. 2 shows the operation of passing from 1,000 rpm to 250 rpm, followed by stopping. This operation takes place when the elevator comes to a stop. /24

c. The brake electromagnet produces sound pulses of approximately 10 dB. What these pulses look like incident to applying (A) and releasing (D) the brake is shown in Fig. 3.



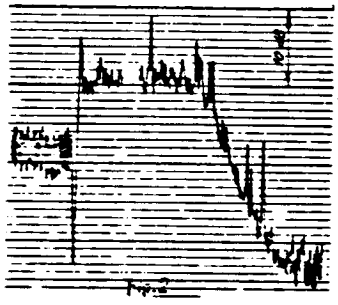


Fig. 2.

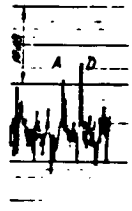


Fig. 3.

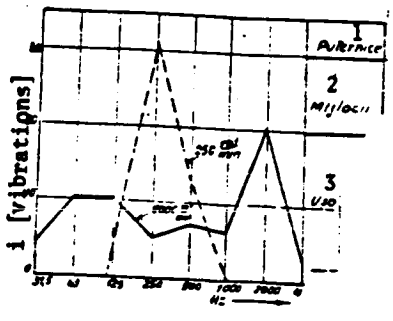
d. Research has shown that the crib for damping vibrations between winch and floor yields good results, the intensity level of the vibrations not being high. This can be seen from Fig. 4.

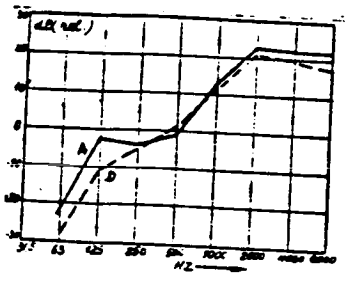
Contactors (25 A, type A) constitute the elevator's control elements. Fig. 5 shows how the noise level of a contactor incident to applying (A) and releasing (D) the brake differs from the Cz65 curve. /24

3. Research on noise sources was complemented with "in situ" research in a test building in which, in order to improve the transmissibility of elevator noise and vibrations, a number of measures were taken, such as:

-- complete insulation of the floor and walls of the winch chamber from the concrete block on which the winch was mounted by using a crib layer, avoiding any rigid connection between concrete block and floor;

-- the holes for passage of the cables to the elevator shaft were very much reduced and lined with plastic tubes interrupted by the crib layer;





-- the control panel with the contactors was mounted in a box that could be sealed with a cover. The box was fastened to the floor on an insulating crib layer;

-- the walls of the winch chamber were soundproofed with plates of FA 140/5 mineral wadding attached directly to the plastering, and the door was lined;

-- the cabin's mobile command cam was redesigned and the cabin's system of double doors was replaced by sliding doors;

-- the elevator shaft was built with care, the walls were given non-floating plastering and the verticallity and parallelism of the cabin's guide rails were respected. The entire outfit was mounted under strict control. /242

4. The efficiency of the measures adopted is illustrated by the results of the noise and vibration measurements conducted after their application.

Thus, Fig. 6 shows the high spectrograms obtained for applying and releasing the contacts before (A, D) and after (a, d) mounting the cover, and Fig. 7 shows the attenuation of the noise level obtained for applying (A) and releasing (D) them after mounting the cover. The efficiency of the solution adopted can easily be seen, especially at high frequencies, where the noise intensity level is reduced by approximately 20 dB.

The results of soundproofing the winch chamber are shown in Fig. 8.

Owing to the introduction of the crib layer and the design and mounting measures adopted, the good vibration damping between

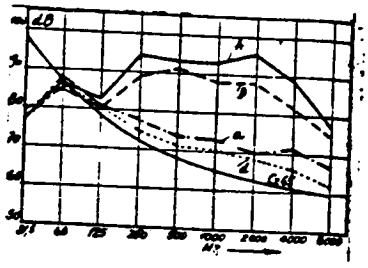


Fig. 6.

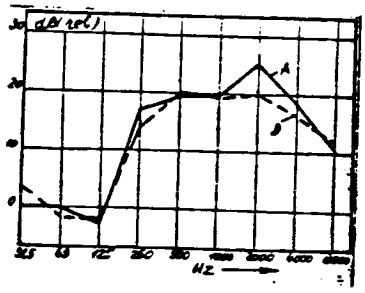


Fig. 7.

concrete block (D) on which the winch is mounted and floor (P) can be seen (Fig. 9).

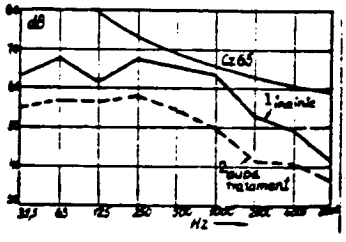


Fig. 8. 1. Before;
2. After treatment.

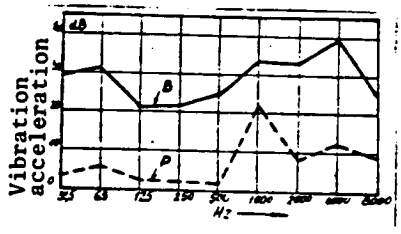


Fig. 9.

5. The results obtained show that the level of elevator noise and vibrations in apartment buildings can be considerably reduced by slight and inexpensive changes in construction. Everything must be carefully and correctly mounted. /2

Noise and vibrations can be still more reduced by direct action on noise sources.

Summary

The authors present the results of experimental research on the reduction of the level of elevator noise and vibrations in apartment buildings. /24

By improving the mounting and gearing conditions of the winch and soundproofing the winch chamber, as well as by covering the elevator's control panel, the noise and vibration level was appreciably reduced.

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REDUCTION OF NOISE GENERATED BY AIR CONDITIONING AND VENTILATION PLANTS AND TRANSMITTED TO INHABITED AREAS

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1. Generalities

In some sectors of the textile industry, especially in weaving /24
and spinning mills and in knit wear and confection departments, technological and labor protection demands call for the observance of special microclimatic conditions, such as maintenance of the temperature's relative humidity values and reduction of the content of harmful substances to concentrations prescribed by the Republican Labor Protection Norms.

In order to meet these conditions, the above-mentioned enterprises are equipped with large-capacity air conditioning plants as well as special mechanical ventilation plants. These conditioning plants are equipped with axial fans whose unit discharges attain 250,000-300,000 m^3/h and whose economical building necessitates adoption of high values for peripheral speeds, which generates noise levels that can reach 100-105 dB/A.

As a result, under some circumstances and especially in the case of enterprises located in the neighborhood of inhabited areas, it is necessary to adopt adequate measures for attenuating the noise propagated to the outside from the conditioning plants, so as to ensure observance of the legislation concerning protection of the environment (Cz45 by day and Cz35 by night).

2. Determinations for Detecting Noise Sources and the Level of Sound Transmitted to Inhabited Areas

In view of this situation, the noise and vibration group of the /24
Light Industrial Design Institute conducted a series of measurements in order to determine the noise level in ventilation and conditioning plants as well as in nearby inhabited areas.

To this end, a number of textile enterprises were analyzed. In the present study we shall discuss findings and measurements as well as some results for three textile enterprises: a silk mill located near an inhabited area; a spinning mill located in an inhabited area; and a stocking manufactory located near an area inhabited by the manufactory employees.

Measurements were recorded with a portable Brüel & Kjaer phonometer equipped with an octave band filter. The spectrograms of these measurements are included in the present study. Analysis of these measurements leads in principal to the following results:

a) The axial fans with which the silk mill and the spinning mill are equipped have discharges ranging from 110,000 to 275,000 m^3/h at a pressure of 40-50 mm H_2O. The level of the noise generated within the factories by these fans are high (100-106 dB/A).

b) As can be seen from the spectrogram of Fig. 1, the level of the noise generated by the conditioning plant fans of the silk mill transmitted through the air intakes to the inhabited area outside at a distance of about 40-50 m from the intakes, exceeds the admissible level set for these zones.

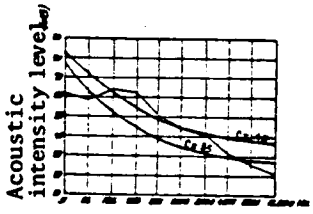

Fig. 1. Noise spectrogram for the conditioning plant fans of the silk mill, transmitted to the inhabited area outside.

c) As can be seen from the spectrogram of Fig. 2 (curve a) the level of the noise generated by the conditioning plant fans of the spinning mill transmitted through the air intakes to the inhabited area outside at a difference of about 8-10 m from the intakes, exceeds by far the admissible level for inhabited regions. /24

d) Each of the departments located on the five floors of the stocking manufactory have two individual conditioning devices with unit discharger of 15,000 m^3/h. The devices are equipped with air intakes placed on the walls facing the inhabited area.

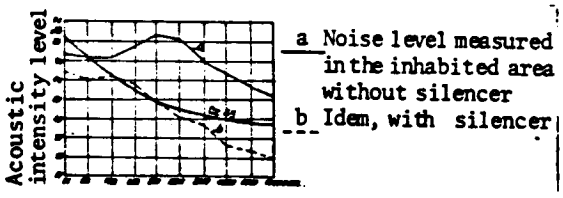

Fig. 2. Noise spectrogram for conditioning plant fans before and after application of silencers.

e) As can be seen from the spectrogram of Fig. 3, the level of the noise generated by these devices transmitted through the air intakes to the inhabited area outside at a distance of 8-10 m from the walls of the factory on which the intakes are placed, exceeds the admissible level.

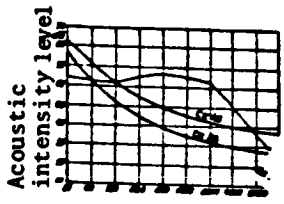

Fig. 3. Noise spectrogram for conditioning devices of the manufacturing departments of the stocking manufactory, transmitted to the inhabited area outside.

3. Solutions for Reducing Noise Transmitted to Inhabited Regions Outside /24

In order to reduce the level of the noise generated by the ventilation and conditioning plant fans and the individual conditioning devices that is transmitted to inhabited areas outside down to admissible levels for these areas, we adopted the following solutions:

For existing units:

a) Soundproofing of rooms located between fans and air intakes (in the case of conditioning and ventilation plants).

b) Placement of active lamellar silencers on the air intakes.

For new units that are being planned:

a) Choice of a type of fan that can be placed in the region of maximum productivity so that the most silent operation can be ensured.

b) Elastic placement of the fans by means of shock absorbers.

c) When new units are placed near inhabited zones, the probable noise level of the fans will be determined beforehand and then, depending on this level, the minimum distance for the noise level will be determined so as to fall within the admissible limits for inhabited areas.

d) If for certain reasons it should not be possible to observe the distance from inhabited regions as established by the method indicated under paragraph c), provision will be made for the designing of noise silencers that will be dimensioned according to the level of noise generated by the source, the distance between air intakes and inhabited regions and the admissible noise level for the inhabited region under the most unfavorable circumstances (at night).

In order to come to the assistance of the design and execution engineers and fitters the Institute for Light Industrial Research worked out two types of silencers: the R.1 with a lamella thickness of 100 mm and the R.2 with a lamella thickness of 200 mm. Lamella length is 1 m, the distance between lamellae is 100, 150 and 200 mm (Fig. 4). These types of silencers were /24

manufactured and tested on the INCERC research stand, and the attenuations that could be obtained with these silencers as a function of lamella thickness and distance between lamellae were determined (Figs. 5 and 6).

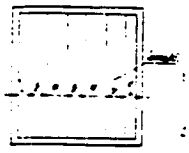

Fig. 4. Active lamella silencer of the R1 and R2 types. g -- lamella thickness; d -- distance between lamellae.
1. Lamella.

Fig. 5. Noise attenuations obtained with R1 active lamella silencers (lamella thickness = 100 mm).
1. Distance between lamellae.

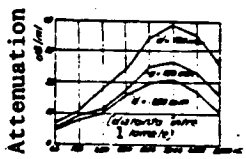

Fig. 6. Noise attenuations obtained with R2 active lamella silencers (lamella thickness = 200 mm).
1. Distance between lamellae.

The main elements of the silencers, namely, the lamellae, were standardized in the sense that on the basis of six model lamellae 48 lamellae were produced with sizes ranging from 250 x 500 mm to 2,000 x 2,000 mm, so that a wide range of silencers could be produced that cover all sections of air passage as a function of mounting positions (in pipes or on air intakes).

4. Efficiency of Noise Reduction Solutions

For the silk mill and the spinning mill noise reduction projects were drawn up.

Up to the present time only the noise reduction project for the conditioning plant of the spinning mill has reached the experimental stage. Noise silencers were applied only on the outside (because these had the greatest acoustic efficiency).

The mineral felt lamellae of these silencers were 100 mm thick and 100 mm apart. The length of the silencer as established by calculation was 2 m.

After doing the necessary work, we conducted noise measurements that showed that applying the above-mentioned silencers the noise level was brought down to the immediate vicinity of the Cz45 curve showing the admissible level for inhabited regions by day (see curve b in the spectrogram of Fig. 2). /2

Conclusions

a) Large-capacity (100,000-300,000 m^3/h)axial fans with which the ventilation and conditioning plants of large textile mills are equipped, generate noise with a high level (100-105 dB/A). The individual conditioning devices used in some confection and knit wear rooms also generate relatively high noise (80-85 dB).

b) In the case of existing enterprises located near inhabited regions, with air intakes facing these regions, the level of the noise generated by conditioning and ventilation plants as well as by individual conditioning devices that is transmitted to the inhabited region, exceeds the admissible level for these regions.

c) By soundproofing the conditioning plant rooms and, especially, by applying silencers of the appropriate size the level of the noise transmitted to the inhabited regions can be brought down to admissible levels.

d) In designing textile enterprises located near inhabited regions steps must be taken to avoid noise pollution of these regions. To this end fans will be chosen for optimum and most silent operating conditions and will be equipped with noise silencers dimensioned in accordance with local conditions.

Summary

The fans with which the conditioning and ventilation plants /25
of weaving and spinning mills are equipped and the conditioning devices used in certain confection and knit wear departments of the textile industry generate loud noise.

Measurements conducted in some existing enterprises placed near inhabited regions have brought to light that the noise transmitted through air intakes to the inhabited regions exceeds the admissible level for these regions.

The authors present solutions for reducing the noise generated by the fans of ventilation and conditioning plants and transmitted to inhabited regions down to the admissible level, as well as the results obtained by experimental application of some noise reduction solutions in the conditioning plants of a spinning mill.

N74-28394

CHARTING ENVIRONMENTAL POLLUTION

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Noise Pollution of the Environment

"Noise is no longer an occupational hazard but has become a threat to the physical and mental health of the general public." Thus we read in the World Health Organization's Cahiers de la Sante Publique No. 30/67. /25

Research has shown that noise pollution affects a disproportionately large area of agglomerated perimeters throughout the world as well as in the territory that constituted the object of the respective studies. For example, a survey conducted in England establishes that about half of city dwellers are affected by noises coming from outside the dwelling (road, aerial and railway traffic) and another 16% suffer because of industrial noise penetrating into residences. As a consequence, it has been recommended that locality systematization projects be proceded by noise studies. Specialists in the field, such as Furrer (Switzerland), Josse (France), Eichler (German Democratic Republic), Joffe (USSR), Close (USA) have stressed the necessity of isolating zones with quietness requirements from noisy zones as early as the town planning stage: the degree of disturbance caused by environmental noise must be considered one of the decisive factors in choosing the location.

Charting the Acoustic Noxa

Noise measurements were conducted at a great number of representative points for the different categories of urban and rural environment and pleasure grounds, following the methodology established by the national norms and the recommendations of the ISO. /25

Knowing the great importance of the parameters of fluctuation in time of noise, such as the ratio of peak and mean values of noise levels, the number of the events with intense noise per unit of time, the nature of the increase (abrupt or slow) of the noise peaks, as well as their distribution in terms of frequency bands, we proceded to make continuous recordings at points of particular importance. The noise was recorded on magnetic tape and then passed through one-octave wide band filters and fixed on paper tape (Figs. 1-3).

Fig. 1 shows the variation of noise in a street affected by industry. It can be seen that the levels remain within rather close limits.

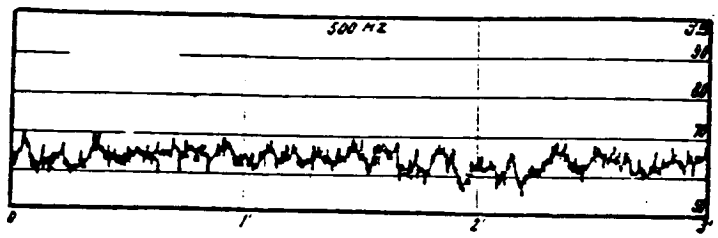

Fig. 1.

Fig. 2 shows the noise recording for an intensely traveled street. In the course of the three minutes corresponding to the recording three peaks can be observed with fluctuations of about 30 dB in the frequency band with its center at 125 Hz. In the 1,000 Hz band (see Fig. 3) the fluctuations exceed 30 dB and have a different shape. While the event at the middle of the second minute increases slowly in time so that it causes a relatively supportable disturbance, the impulsive noises of the first minute and of the beginning of the third minute have the character of serious auditory disturbance. /25

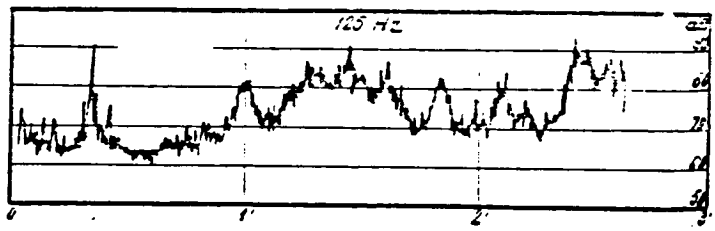

Fig. 2.

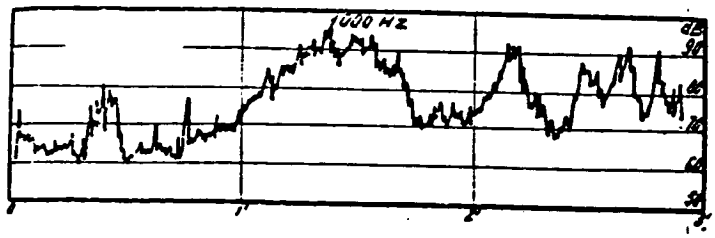

Fig. 3.

The method of continuous selective recording seems to provide more pregnant information about fluctuating noises than their expression by the notion of equivalent noise

$$L_{ech} = \frac{1}{t}\int_0^t L_{(t)}\,dt$$

Zoning

Acoustic measurements recorded according to the above-mentioned methodology were used in zone delimitation in order to permit town planning designers to take into consideration environmental factors. As a result, we applied the principle that the order of accuracy of the delimitations be about 100 m inasmuch as greater accuracy is not significant in town planning.

In the course of the studies the conclusion was reached that /25
the two categories "silent zone" and "noisy zone" are not sufficient. It was necessary to introduce the concept of mixed zone, in which sources of intense noise are located close to buildings that must be protected.

In what follows we shall attempt a succinct characterization of the proposed zoning.

Silent Zone

In this zone any type of social-cultural building can be erected and, in the first instance, residences. As noise sources, apart from those connected with the construction of the proposed building itself, we might mention traffic, delivery of goods, services and utilities.

Mixed Zone

This is usually the result of uncontrolled development. For town planning purposes the mixed zone must gradually be transformed into a silent zone; in those cases when this desideratum cannot be realized, for example, in commercial and administrative quarters, in those with a historic character or architectural monuments, the placement of new buildings will be preceded by analysis of neighborhoods, taking the necessary steps for reducing the degree of reciprocal disturbance. In principle, the mixed zone is not suitable for residences but is suitable for social-cultural purposes such as sports buildings and institutions of learning and health other than hospitals and sanitariums.

Noisy Zone

This is characterized by noise levels that exceed the sanitary norms and are earmarked for industry and intense railway, road and air traffic. Included in the noisy zone are reserved spaces for future industrial buildings, as well as protective bands that accompany highways. Let us note that on the basis of the determinations conducted for the present study, these protective bands have a width of 2 x 150 m for intensely traveled streets, which is the same width as for protection (in open country) against exhaust gases. If the noisy zone cannot be restricted as to area, it is desirable that steps be taken to diminish the polluting influence by reducing the noise energy radiated by the different /25
units found in the zone. Let us mention, by way of example, the case of some Cluj productive units that are located at the edge of industrial zones and would not normally have to constitute sources of noise pollution because the noise does not come from the basic technological process itself. Thus, two fans subsequently mounted on the roof of one of the sheds of the Baciu grist mill pollutes residential quarters and Hoia pleasure park for a radius of 2.5 km. Similarly, the TCC production center situated in Cluj on Quietness Street (!!) pollutes for a radius of about 2 km.

Conclusions

A study concluded on the territory of an industrial district (Cluj) compared with another territory located at the edge of industrialization (Salaj) provides useful data for the drawing up of town planning studies and particulars and criteria for the comparison and optimization of building sites.

For the present state of the art of town planning, dividing the territory into three zones (silent, mixed and noisy) seems to be sufficient and satisfactory.

Moreover, it has been found that, apart from a few exceptions, areas affected by different polluting agents are within the limits traced for high noise level areas; consequently, we propose that high acoustic pressure levels be regarded as the primary indication of environmental pollution.

Summary

The authors present a complex methodology for charting environmental pollution due to physical, chemical and biological noxious agents on the scale of an industrial district. /25

The paper offers data for gradual reducing of intensely polluted areas as well as for the demarcation and pollution control of sites meant for dwelling and similar purposes.

It is found that the areas affected by different noxious agents are, apart from a few exceptions, within the limits traced for high noise level areas; consequently, it may be suggested that high noise pressure levels should be used as the primary indication of environmental pollution.

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N74-28395

AN OBJECTIVE METHOD AND MEASURING EQUIPMENT FOR NOISE CONTROL AND ACOUSTIC DIAGNOSTICS OF MOTOCARS

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General description

Many methods and apparat for evaluation of noise produced by motorcar engines have been used till now. They consist, generally, in measuring either the overall noise level in the whole audio-frequency range or the sound pressure levels in s bitrarily chosen frequency bands. These methods, however, being very useful in comparing the noisiness of the vehicle under test with appropriate standards, do not give any information concerning the actual technical condition of the object and do not provide facilities for localizing the source of the excessive noise which may be attributed to defects in various parts and assemblies of the engine.

To avoid these difficulties we have proposed an objective accustic method and designed an universal measuring equipment which provides facilities for quick and direct estimation of the behaviour of the engine at definite running speed, for measuring the noisiness of the engine and its particular assemblies, for identification and localization of faults and defects and for comparing the working parameters of the engine under test with corresponding standards.

To this effect the apparatus has been equipped with two sets of low-pass and high-pass filters, respectively, whose pass-bands are so adjusted as to cover the most distinctive frequency ranges in which the noise spectra of acoustic phenomena associated with the working conditions of individual assemblies of the engine to be tested at definite running speed are contained. The output of each filter is combined with the amplitude discrimi-

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filter whose threshold level may be individually adjusted according to special program which describes the acoustical characteristics of the standard engine representing the considered type or model. The apparatus is equipped with a conventional electronic revolution indicator or counter so that the results of acoustic measurements may be directly attributed to the arbitrarily chosen working speed of the engine.

The pass-bands of the low-pass filter set correspond to those frequency ranges in which acoustic phenomena occur which are associated either with the work of the shifting assemblies and mechanisms such as pistons and valves or with secondary effects caused by the ignition and detonation processes and by the flow of combustion gases through the exhaust system and pipe. It should be mentioned that at least three lowest harmonics of the fundamental running frequency must be located within the low-pass filter frequency range.

The pass-bands of the high-pass filter set on the other hand cover those frequency ranges in which acoustic phenomena may be observed which are caused either by the turbulent effects associated with suction and exhaustion processes or by noise due to the work of rotating assemblies such as bearings, gears, fans etc.

The choice of two above mentioned frequency ranges has been motivated by the results of experiments carried on passenger cars and trucks of home and foreign production [1]. It has been proved that the noise spectra of engines with spark ignition are concentrated within the low frequency range, viz. below 500 cps and may be attributed to exhaustion processes mainly which are responsible for the overall noise level. Noise spectra of engines with self-ignition are much broader and more uniform and have only a few local maxima in the low frequency range, especially if the exhaust system is out of order, see Fig.1.

Similar effect may be observed in Fig.2 which represents the noise spectra of two-stroke and four-stroke engines, respectively. As it may be seen, the noise spectra of two-stroke engines are much broader than those of four-stroke ones which decrease at the rate of about 6 db/octave above 500 cps. In this connection the most frequently used value of the cut-off frequency of the high-pass and low-pass filter sets is equal to 500 cps.

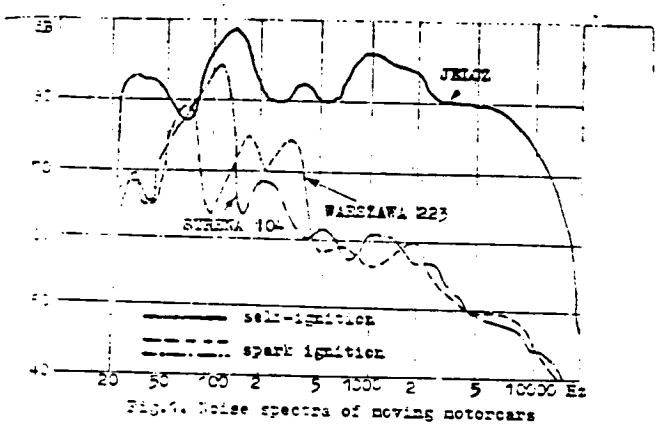

Fig.1. Noise spectra of moving motorcars with spark ignition and self-ignition engines, measured in 1/3-octave bands

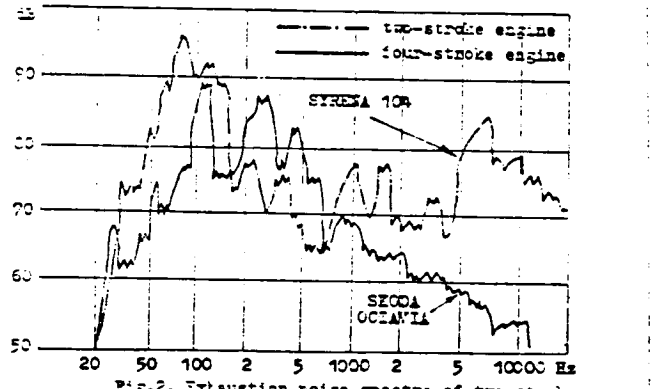

Fig.2. Exhaustion noise spectra of two-stroke and four-stroke motorcar engines in 1/3-octave bands

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...ational information concerning the mean values of exhaust ... levels and predominant frequency ranges of exhaustion ... of various types of motorcars are given in Table 1.

... ...ean values of exhaustion noise levels measured ... ...istance 1 m. from the exhaust pipe and predo... ...frequency ranges of exhaustion noise spectra of ... ...types of motorcars. The running speed is equal ... ... values for spark ignition engines and 4/4 max. ... ...-ignition engines, respectively.

...es of vehicle	Overall noise level dB_A	Predominant frequency ranges of noise spectra /cps/
... of ... construction	84,5	100–300; 800–1600
... ...s of ...-... producer.	83,5	50–300; 1000–2000
..., weight ... 1,5 t	81,0	50–80; 1500–2200
..., weight > 2,5 t	84,2	50–160; 1000–2200
... ..., self-ignition	92,7	50–300; 1000–2000

Instrumentation

The ... diagram of the measuring equipment adopted to ... ...of internal combustion engines, both two- ... ...-stroke ones, with spark- or self-acting ignition, ... The gauge 3 of the electronic revolution counter is connected to the ignition distributor or to the spark ... of the engine under test and the actual value of the working ... ...can be read on the indicator's scale. The standard condenser microphone M is placed at the appropriate point in the space ... ...the vehicle T, according to the actual noise control ... ...and recommendations [2], [3]. The electrical output ... of the microphone pre-amplifier PA is led to the input ... the sound level meter SLM and to the inputs of low-pass /LPF/ ... high-pass /HPF/ filters being connected in parallel.

NOT REPRODUCIBLE

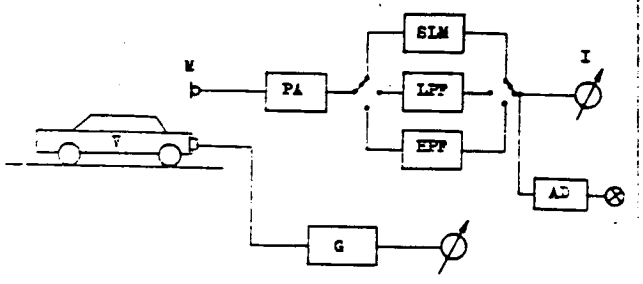

Fig.3. Block diagram of the measuring equipment for noise control and accustic diagnostics of cars

All filters are of the active type using integrated circuits MAA 702 and their cut-off frequencies may be adjusted in broad limits according to the nominal revolution speed of the engine to be tested. Most frequently used cut-off frequency is 500 cps. The output of the sound level meter as well as the outputs of individual LP and EP filters may be read on the scale of the indicating instrument I which is calibrated in dB_A for sound level measurement and in dB re $2 \cdot 10^{-5}$ N/m^2 for band pressure level measurement.

The apparatus is equipped with amplitude discriminator AD whose threshold level is adjustable fluently and in 5 dB steps. Whenever the previously adjusted threshold level have been exceeded in any frequency band, the self-sustaining electronic device starts and the control signal is sent which may be switched off by pushing the respective cancelling button.

The measuring equipment as a whole is battery operated and easily transportable, its total weight does not exceed 4 kG and thus it may be used in any environment, open-air conditions in-

cluding. The admissible temperature range is contained within the limits from -5° C to +40° C. In general, the apparatus is adapted for use either on the production control line in the factory or in the temporary testing of motorcars on diagnostic stands. In the simplified performance the apparatus may be used by the private owner of the motorcar, too.

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Summary

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The subject of the present report is an objective method and apparatus for noise control and acoustic diagnostics of motorcar engines. The method and apparatus let us know whether the noisiness of the vehicle under test exceeds or not the admissible threshold levels given by appropriate standards and if so what is the main source of the excessive noise. The method consists in measuring both the overall noise level and the sound pressure levels in definite frequency bands while the engine speed is controlled as well and may be fixed at prescribed values. Whenever the individually adjusted threshold level has been exceeded in any frequency band, a self-sustaining control signal is sent. The apparatus may be used on the production control line in the factory or in temporary testing of motorcars on diagnostic stands.

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N74-28396

CONCERNS OF THE INSTITUTE OF TRANSPORT STUDY AND RESEARCH FOR REDUCING THE SOUND LEVEL INSIDE COMPLETELY REPAIRED BUSES

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The efforts of the Institute of Transport Study and Research /269
to reduce the acoustic pressure level inside completely repaired buses has been motivated by the permanent desire to increase the comfort of riders during transport and to create better conditions for bus drivers. Noise and vibrations produce effects that are harmful for man's health and have a negative influence on his productivity.

In order to control noise inside buses the noise sources must be known and then each must be treated separately.

The total noise of a bus is made up of many simultaneously acting sound sources. These sources differ very much in their manner of emission as well as in the intensity and shape of their general noise spectrum. For this reason, precise delimitation of the physical characteristics of each source separately raises difficulties.

The most important noise sources that appear during a bus ride are due to the running of the motor, the transmission, the brakes, aerodynamic resistance and the bus's rolling.

The total noise generated by these moving parts and the vibrations of the bus body have a harmful physiological as well as psychological effect on the bus driver as well as on the rider. For this reason it is necessary that the interior of the bus be soundproofed.

The noises produced by these sources cover a very wide range of frequencies, in the audible region (20-16,000 Hz) as well as in the ultrasonic region, and have a continuous spectrum throughout the frequency range. /270

The shape of the frequency spectrum of a noise is determined through analysis with the aid of special acoustic apparatus. The acoustic level was measured according to the existing methodology adopted from the CAER norms.

In order to attenuate the acoustic pressure level the noise inside the bus must be analyzed at different speeds and for different types of road.

The noise inside buses was analyzed by:

-- measurement of the acoustic pressure level in frequency ranges (the spectral characteristic of the noise);

-- measurement of the acoustic intensity level.

This detailed analysis of noises is necessary because the efficiency of their damping varies with the acoustic frequency and intensity. Those sound sources must be damped which generate noises with the highest intensity and at the most annoying frequencies (for example, 3,000 Hz).

Our subject of research was the problem of damping the acoustic pressure inside completely repaired buses. In order to solve this problem our research team, after analysis of the sound spectrum of completely repaired buses that had not been soundproofed, studied possible soundproofing methods to be applied for complete repair. Thus, it was found that the acoustic pressure level could be damped by:

-- soundproofing the body with a 2-3 mm thick layer applied on the inside of the lateral and frontal walls, the roof and the floor;

-- lining the hood with:

· a 2-3 mm thick layer of series 1,200 antinoise paint;

· a 10 mm thick layer of mineral wadding felt;

· a panel made of perforated aluminum plate attached directly to the partition separating the motor compartment from the rest of the bus;

· parts inside the bus must be firmly attached in order not to cause additional noises. /27

Our methodology in measuring the acoustic pressure level was as follows:

-- surveying of the spectral characteristic of the acoustic pressure level inside completely repaired buses in the following cases:

· unsoundproofed bus;

· bus soundproofed with a layer of antinoise paint and sealing off the motor compartment from the rest of the bus;

· bus soundproofed by lining the inside with melamine plates and upholstered seats;

-- the acoustic pressure level was measured under identical conditions of the highway, the speed of the bus and the number of passengers;

-- the acoustic pressure level inside the bus was measured in the following regions: beside the driver (CA); in the middle of the bus (m); at the rear of the bus, beside the motor (M).

Before discussing the results we obtained in damping the acoustic pressure level inside buses, we must show that we tried to fit the level after soundproofing into harmless noise zone C or B as specified by the ISC and STAS norms.

The spectral characteristics of the acoustic pressure level inside completely repaired buses at a speed of 60 km/h are shown in Figs. 1, 2 and 3.

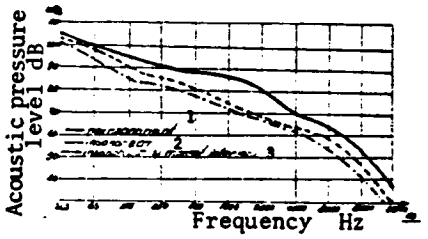

Fig. 1. Spectral characteristic of acoustic pressure level inside al-OL-2491 bus at 60 km/h (measured near driver). 1. Unsoundproofed; 2. Soundproofed; 3. Soundproofed and treated interior.

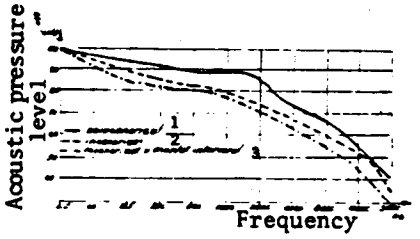

Fig. 2. Spectral characteristic of acoustic pressure level inside al-OL-2491 bus at 60 km/h (measured in middle of bus). 1. Unsoundproofed; 2. Soundproofed; 3. Soundproofed and treated interior.

Fig. 4 shows in percents the attenuation of the acoustic pressure level inside the bus as a function of the frequency range.

The influence of the motor's hood was analyzed by means of measurements of the spectral characteristics of the acoustic pressure level as a function of the speed of the motor with the hood open or closed (Fig. 5). /27

Our research team compared the attenuation of the acoustic pressure level due to the hood in the following buses: competely repaired TV 20; standard model TV 20; prototype TV 20; SETRA and Pegaso. The curves plotted on the basis of these measurements are shown in Fig. 6. The attenuation of the acoustic pressure level obtained by means of the above-mentioned soundproofing measures and be seen from the Figures:
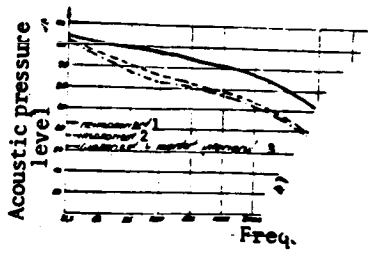
Fig. 3. Spectral characteristic of acoustic pressure level inside a1-OL-2491 bus at 60 km/h (measured at rear seats). 1. Unsoundproofed; 2. Soundproofed; 3. Soundproofed and treated interior.

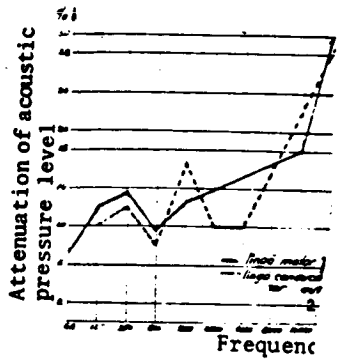

Fig. 4. Attenuation of acoustic pressure level inside bus at 60 km/h. 1. Next to motor; 2. Next to driver.

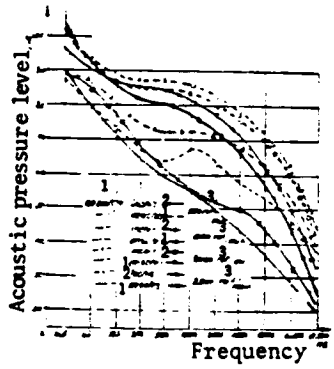

Fig. 5. Spectral characteristic of the acoustic pressure level inside a1-OL-2491 bus for different motor speeds (stationary). 1. Hood; 2. Closed; 3. Open; 4. rpm.

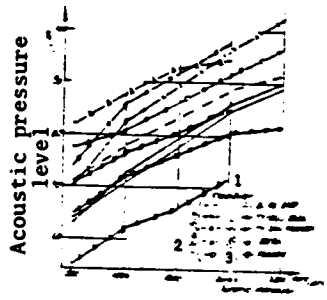

Fig. 6. Acoustic pressure level in case of closed hood (solid line) and open hood (dotted line) as a function of motor speed. 1. Hood; 2. Open; 3. Closed.

-- attenuation of the acoustic intensity level when the hood is open or closed (Fig. 7);

-- attenuation of the acoustic intensity level as a function of the measurement site and the speed of the bus on a road of category I (Fig. 8);

-- attenuation of the acoustic pressure level in percents as compared with the initial level as a function of the measurement site or the speed of the motor (Figs. 9 and 10). /2

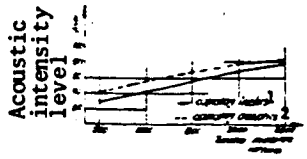

Fig. 7. Acoustic intensity level inside a1-OL-2491 bus in the case of closed and open hood. 1. Closed hood; 2. Open hood.

Fig. 8. Acoustic intensity level inside a1-OL-2491 bus on a road of category I. 1. Phones; 2. Unsoundproofed; 3. Soundproofed.

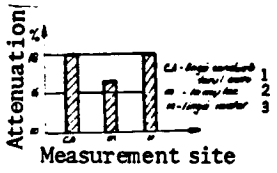

Fig. 9. Attenuation of the acoustic pressure level inside completely repaired buses. 1. Near driver; 2. Middle; 3. Near motor.

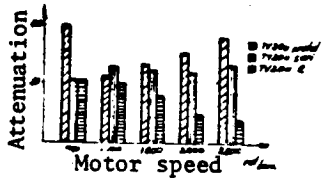

Fig. 10. Attenuation of acoustic pressure level due to hood.

From the foregoing it follows that by soundproofing the body of a completely repaired TV 20 bus with antinoise paint and providing a correct hood, the acoustic pressure level can be reduced by:

-- over 10% thanks to the hood for motor speeds of 500 and 1,000 rpm and by about 5% for other speeds;

-- 18% near the driver and the motor and 15.5% in the middle of the bus;

-- 7-30% is the attenuation of the acoustic pressure level in the frequency range. It can be observed that the greatest attenuation is for the higher frequencies between 1,000 and 31,500 Hz (Fig. 4).

It can also be shown that the attenuation of the acoustic pressure level in the case of the completely repaired buses that our research group tested is better than for the standard model of the TV 20 bus and the Pegaso.

The acoustic pressure level inside a completely repaired bus at legal speeds belongs in the harmless noise region, so that /274

better transportation conditions have been created for riders and better working conditions for drivers.

Summary

In recent years much importance has been attached to noise control of buses in order to increase passenger and driver comfort and reduce the noise generated by buses as a noise source in the environment. /275

The authors describe the efforts of the Institute of Transport Study and Research to measure the sound level of the noise sources on buses and to attenuate the acoustic pressure level inside the bus by using soundproofing means that can be applied during complete repair.

In the course of their study the authors conduct a spectral analysis of the sound level as a function of the speed of the motor, the speed of the bus along the road and the category of road.

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N74-28397

SOME ASPECTS OF SOUND DISCOMFORT CAUSED BY STREETCAR TRAFFIC

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Timisoara Institute of Medicine
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The noise in large conurbations necessitates that noise sources and their characteristics be known in order to ensure efficient protection of the organ of hearing exposed to this "noise pollution". /27

Among urban means of conveyance the streetcar constitutes the most important noise-generating source. We measured the intensity of streetcar noise on different lines and inside different types of streetcars of the town of Timisoara.

Research Methods Utilized

Noise intensity was measured with a Brüel & Kjaer 3501 sonometer when different types of streetcars arrived and came to a halt at streetcar stops. We also measured noise intensity inside different types of streetcars. Measurements were conducted for total noise as well as in the region of the A, B, C curves.

Research Results

Noise intensity measurement at the streetcar stops of the common 1-2 line (Gara Timisoara-nord [North Timisoara Station] -- Timisoara-est [East Timisoara Station]) disclosed in the case of streetcars of the old type intense noise that sometimes reached 97 dB.

Measurement of the noise generated by the new silent type of ITT streetcar on the same line brought to light a lower noise intensity. The results are compared in Figs. 1-8 (the hatched parts of the columns indicate the values recorded for the new ITT streetcars).

The results of noise intensity measurement in some noisier zones are shown in Fig. 9. /27

Noise intensity measurement inside streetcars of the old type brought to light the values shown in Fig. 10.

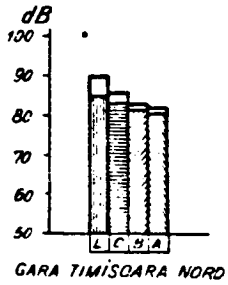

Fig.1.

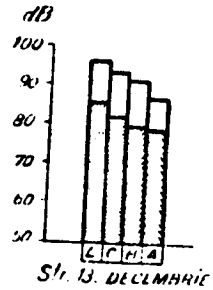

Fig.2.

Fig.4.

Fig.5.

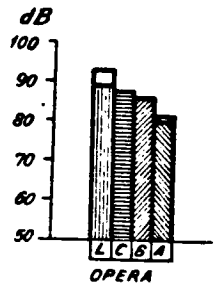

Fig.6.

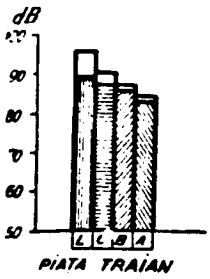

Fig.7.

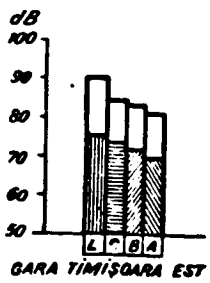

Fig.8.

The results of noise intensity measurement in the new silent type of ITT streetcar are shown in Figs. 11-12.

Discussion of Results and Conclusions /281

Streetcar noise constitutes an important factor of discomfort for the population, reaching and sometimes exceeding the noise levels encountered in production.

Noise generation is more particularly due to rolling in itself, the transmission system and the brake. The noise inside streetcars is the result of the vibrations and slamming of different inner and

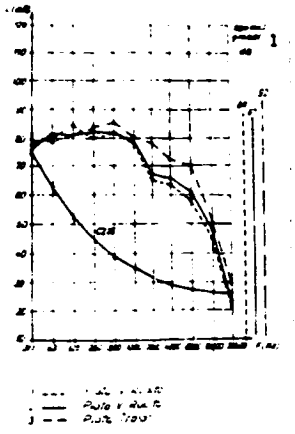

Fig. 9. 1. Total noise.

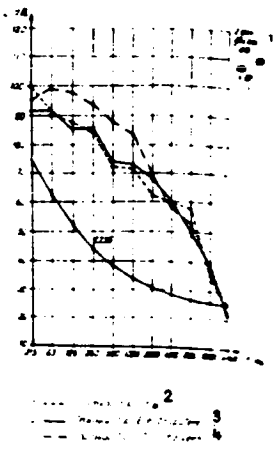

Fig. 10. 1. Total noise; 2. Streetcar of the ITB type; 3. Streetcar of the "electroputere" type; 4. Streetcar of the ITT type (old type).

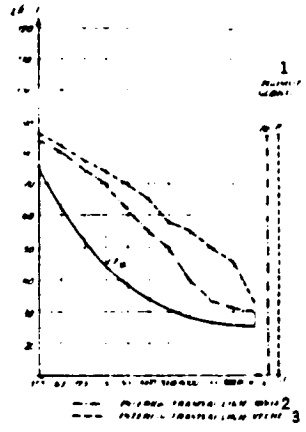

Fig. 11. 1. Total noise; 2. Inside streetcar of new line; 3. Inside streetcar of old line.

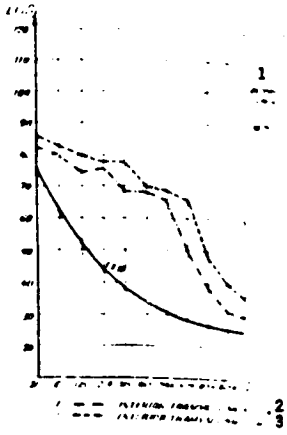

Fig. 12. 1. Total noise; 2. Inside streetcar of new line (redone); 3. Inside streetcar of old line.

outer component parts, the rattling of the wheels due to nonuniformity of the rails, the undulatory wear of the rails in the form of periodic dislevelments along the rolling surface of the rail end, etc.

Also involved in the generation of external noise are running gear shocks, brake shoes, braces, carriage walls, etc.

Measurements conducted in carriages of the same type and under identical conditions have disclosed different noise intensities. This is due to such factors as the different technological state of the carriages, the traveling speed, braking on bridges, peak traffic, the different undulatory wear of the rails, etc.

For reducing the intensity of streetcar noise we recommend better soundproofing of the different constructional elements and more silent joining of the different components.

In the case of the new type of silent streetcar manufactured by ITT we found a pronounced reduction in the noise generated at the moment of stopping, at some stations by 15-17 dB. The exposure of passengers to noise is also much more reduced inside the new type of ITT streetcar, sometimes reaching a reduction in noise intensity by 18-20 dB as compared with the noise in old types.

Our results show that streetcar noise acts with a variable intensity by day as well as partially by night, affecting all age groups and exceeding the admissible intensities by 45 dB by day and 35 dB by night.

But in addition to the organ of hearing, which is much more sensitive to the action of noise in childhood, noise pollution of populated centers also affects many other anatomical functional systems of major importance in the economy of the human organism. /28

The general fatigue and nervous stress to which the organism exposed to sound aggression is subjected are due to the structure of the auditory canals and the numerous connections that these have in the organism at all levels. The appearance of intense and unexpected noise perturbs activity, entailing a temporary drop in performance, whether it be a question of predominantly motor or intellectual activity.

The building of streetcars that are as silent as possible will contribute greatly to the creation of a sound comfort that is favorable to the carrying out of human activities as well as to the preservation of health.

Summary

Streetcar noise was investigated on different lines and inside different types of streetcars. Noise measurements were conducted with a Brüel & Kjaer sonometer. /28

The results obtained on streetcar line 1-2 (North Timisoara and East Timisoara Station) showed variations of the total noise intensity between 88-97 dB. In some squares with heavy traffic the total noise intensity reached 106 dB. Noise intensity measurement inside different types of streetcars brought to light high values between 101-106 dB while in the case of the new silent type of streetcar the values were 86-87 dB.

These results demonstrate the importance of the sound discomfort produced by streetcar traffic, inasmuch as the noise intensity exceeds by far the values recommended by communal hygiene.

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N74-28398

DATA ON THE ACOUSTIC COMFORT OF PASSENGERS IN RAILROAD CARS AND SOUNDPROOFING RECOMMENDATIONS

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Transportation is already involved in a revolutionary process /28
of development from the triple standpoint of speed, safety and comfort.

As far as railroad transportation is concerned, the concept of passenger comfort might be defined as the sum of elements that contribute physically or subjectively to the sensation of well-being, safety and pleasure of the passengers.

If we leave aside the time-tables, rapidity and frequency of trains, it can be said that the elements that contribute to passenger comfort in a coach are:

-- access, available space, services;

-- general arrangements, quality of seats, decorations;

-- lighting, ventilation, heating, air conditioning;

-- vibrations and mechanical stresses;

-- soundproofing of compartments (Fig. 1).

As regards acoustic comfort, it can be said that excessive noise is the most disturbing factor, causing additional fatigue, impairing conversation or sleep and increasing anxiety or the impression of danger.

I. Measurement and Criteria for Evaluation of Acoustic Comfort

We had to overcome two kinds of difficulties:

-- objective, because in order to define acoustic comfort it is necessary to have equipment that permits the gathering of one or more numerical values that are accurate enough to be acceptable as representative of noise;

-- subjective, because in all cases the perceiving observer /28
interprets individually, in a personal manner, the effects of noise and so it is very difficult, if not downright impossible,

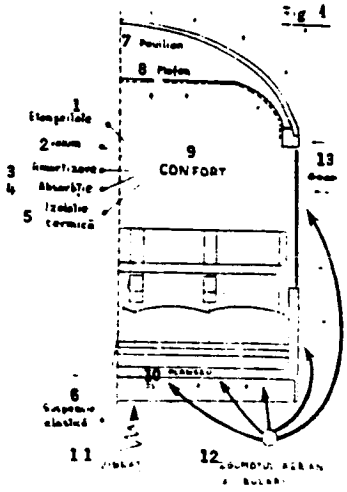


Fig. 1. 1. Tightness; 2. Volume; 3. Damping; 4. Absorption; 5. Thermal insulation; 6. Elastic suspension; 7. Roof; 8. Ceiling; 9. Comfort; 10. Floor; 11. Vibrations; 12. Air noise of rolling; 13. Window pane.

to adopt incontestable criteria for characterizing what is meant by noise-induced discomfort.

The interpretation of results is always delicate because acoustic fields can be free or diffused, noises or sounds can be sustained, fluctuating or impulsive and of different intensities, secondary effects depending on the width of the critical band according to the masking effect, exposure time or conditions of appearance and disappearance.

In order to estimate the acoustic comfort in passenger cars of the Rumanian Railroad we made use of the following representative values (Fig. 2):

-- total noise level in dB (A);

-- spectral analysis in octaves;

-- indices of word intelligibility, according to the studies of Beranek, for a conversation with a normal voice between two interlocuters separated from each other by 0.30 m;

-- comparison of these values with those described by STAS 6661-70.

II. Measuring Methods

Microphone pick-ups were placed under the coach and in the compartments in order to pick up, at different speeds, the noise spectrum and the total level and in order to calculate the indices of intelligibility (i%). We carried out measurements on the axle box, the bogie chassis or the carriage body, properly speaking. Rear and middle compartments of the car were chosen. Compartment doors and windows were shut and the ventilating, heating and lighting systems were put into operation. /28

Our studies were conducted on a track conforming to the recommendations of STAS 6661-70.

The data were gathered on graphic recorders with logarithmic resolution, which yields the evolution of the total level in dB (A).

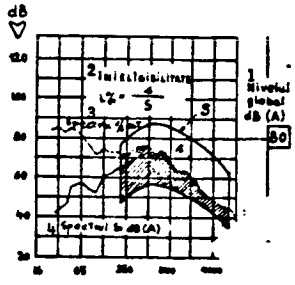
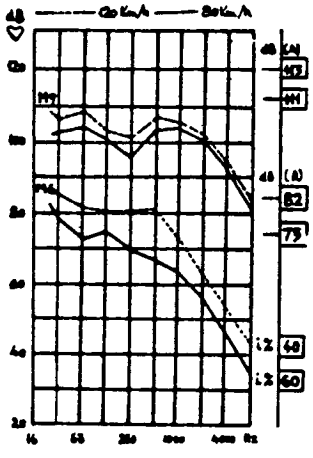


Fig. 2. 1. Total level; 2. Intelligibility; 3. 1/3 oct. spectrum; 4. Spectrum in dB (A).

III. Analysis of Results

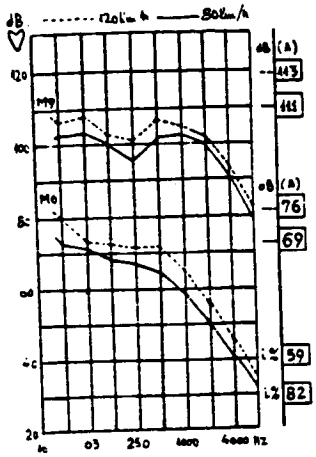
The cars we studied were first-class and second-class passenger cars of Roman construction (Arad Car Plant) making up fast trains and express trains equipped with Minden-Deutz bogies.

The second-class cars from the Arad Car Plant, with ten compartments and 80 seats on settees with a floor of simple design with a bitumen layer, a polystyrene layer and a linoleum covering, with a thickness of about 70 mm and a weight of approximately 30 Kg/m^2, simple 6 mm thick window panes, seats upholstered with plastic, equipped with Minden-Deutz bogies, yielded on an average the following measurement results: /28



Speed Km/h	Compartment			
	Rear		Middle	
	dB(A)	1 %	dB(A)	1 %
120	82	40	79	50
80	75	60	71	73
difference	7	20	8	23

The first-class cars from the Arad Car Plant, with nine compartments and 54 armchair seats, with a floor of simple design with a bitumen layer, a polystyrene layer and a linoleum and marquette covering, with a thickness of about 75 mm and a weight of approximately 35 Kg/m^2, simple 6 mm thick window panes, plush-covered seats, equipped with Minden-Deutz bogies, yielded on an average the following measurement results (M6 and M9):



Speed Km/h	Compartment			
	Rear		Middle	
	dB(A)	1 %	dB(A)	5
80	69	82	64	9
120	76	59	72	1
Difference	7	23	8	8

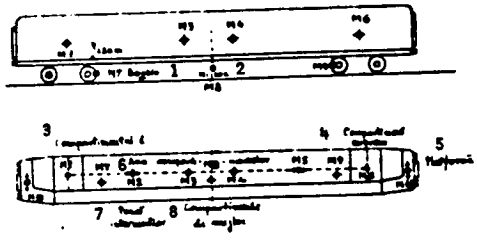


Fig. 3. 1. Bogie; 2. Middle; 3. Compartmental; 4. Rear compartment; 5. Platform; 6. Compartment axis; 7. Intermediate point; 8. Middle compartments.

IV. Discussion of Results and Design Recommendations /289

The principal noise source of all passenger cars is the rolling of steel wheels along steel tracks. The more numerous the shocks and the higher the speed, the greater is this emission. A significant contribution is made by the heating, ventilating and air conditioning plants as well as by the electric power plants. Finally, all vibrations that result from the excitation of design elements through surface resonance, through vibration of thin walls, windows, the bogie chassis and the mechanisms associated with it, must be avoided or stopped in their propagation.

Effective protective measures might be:

-- elimination of vibration transmission to the carriage body;

-- insulation of the spaces subjected to the air noise that envelopes the carriage body;

-- dissipation through absorption of the sound energy that has penetrated into the compartments.

Special attention must be paid to the following:

1. Bogie and bogie-body connections, by suppression of pivots, /29
adoption of silencers, reenforced multilayered rubbers, etc., which will be conducive to breaking "sound bridges" and will, therefore, damp vibrations.

2. The floor must have a continuous structure of its surfaces in order to ensure thermal and acoustic insulation. It is situated closest to the principal noise sources and its composition of different materials whose acoustic impedences combine, must allow reduced sound transparency to be obtained. The first thing to be provided for is the floor's mass, the materials for damping being chosen and combined on the basis of the noise spectrum. Mean damping between 125 and 2,000 Hz must be 50 dB, which presupposes a floor of 35 to 40 Kg/m^2 for a thickness of about 75 mm. Floors with a thickness from 125 to 150 mm, with a mass of 60 Kg/m^2, ensure remarkable phonic and thermal insulation. Floors with greater thicknesses and weights of more than 70 Kg/m^2, with an air space between the layers, ensure a damping of 65 dB. A floating floor permits the interior decoration to be totally separated from the metal of the carriage body, placing the structure of the floor on blocks that are appropriately distributed and insulated by means of elastic elements.

3. Lateral walls, partitions, doors must be lined on the inside with sound-absorbing or sound-insulating materials; partitions must generally have a multilayered structure.

4. It is recommended that windows be double and that they be fit into a single frame, with an air space between the panes.

5. The roof and all ceiling surfaces in the compartments and corridors or platforms must be treated for damping and absorption. We recommend porous perforated materials, made of mineral or organic fibers.

Summary

The definition of optimum passenger comfort brings out the /29
importance of the sound level inside passenger cars.

In order to compare the acoustic comfort of Rumanian Railroad cars we adopted as representative the following values: the total noise level in dB (A), the octave sound spectrum in dB and indices of intelligibility. Microphone pick-ups were placed under the car and in the compartments. Investigations were conducted on a track that meets the requirements of STAS 6661-70.

The noise level perceived inside the car results from two components: one due to the penetration of air noise, and another due to the transmission of vibrations through solids.

Measurement results showed the necessity of improving the bogie and bogie-body connections, intensification of sound-proofing of the floor, adaption of windows with double panes, etc.

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USE OF CONVENTIONAL EQUIPMENT TO MEASURE THE EFFECTIVE PERCEIVED NOISE LEVEL OF AIRCRAFT

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Introduction

The method for measurement of the effective perceived noise level due to an aircraft is defined by ISO [International Organization for Standardization] Recommendation R 507. At first glance this method seems to be applicable only to specific systems of measurement. The measurement equipment is excessively costly and consequently can be acquired only by acoustics departments which do not specialize in the measurement of aircraft noise. The object of the present article is to show that it is nevertheless possible to measure aircraft noise in conformity with ISO Recommendation R 507 by the judicious use of a type of equipment possessed by any acoustics laboratory. /293

To show the justification for this type of method we will first review some basic ideas, as well as the requirements of ISO R 507.

I. Reasons for the Selection of a Specific Method and Unit of Measurement

It is obvious that a reading of the complete spectra of acoustic pressure will permit a perfect description of a sonic field. In all noise problems, however, and especially in the case of the intense noise emitted by aircraft, it is of prime importance to be able to foresee the reactions of human beings subjected to this noise. A unit of measurement has thus been selected which makes it possible to calculate human reactions. This

is why standardization of the method to be used has been based on psychoacoustic research which has defined the subjective values attached to a given noise. The work of Kryter in the 1960's revealed that there are sometimes considerable differences between the sonics of a sound and the discomfort resulting from exposure to it.

Thus in a comparison of two sounds differing in level and frequency, the sound judged to be the louder of the two does not always produce the greater discomfort. Kryter has thus defined the "annoyance" of a sound. This idea is especially appropriate to the measurement of aircraft, since it is precisely the discomfort produced by the sound which it is most important to determine. /294

For this reason aircraft noise will be measured by the method described by Kryter and the measurement will be expressed either as annoyance or as PNdB.

Kryter's work has also shown that if the sound analyzed includes marked irregularities within its spectrum, for example dominant frequencies, a correction must be made in the perceived noise level. A correction must also be made in the perceived noise level to account for the length of exposure to the noise. This method and the corrections to be made are found in ISO Recommendation R 507.

II. The Method for Aircraft Noise Measurement Recommended by ISO R 507

2.1. Acquisition of Data

The measurement system will include a microphone, filters and detector circuits whose dynamics and time constants should correspond to CEI [Commission électrotechnique internationale; International Electrotechnical Commission] publications on precision

sonometers and 1/3 octave filters.

Every 0.5 second and throughout the period of analysis, this system should furnish the sound-pressure level within the 24 1/3-octave bands occurring between 50 Hz and 10,000 Hz. This spectra should be memorized and then processed.

2.2. Processing of Data

Each spectrum read in this way should undergo two stages of processing:

-- calculation of the perceived noise level (PNL);

-- calculation of the correction necessary due to irregularities in the spectrum, termed "pure tone correction."

2.2.1. Calculation of Perceived Noise Level

Given the spectrum of the sound, the partial annoyance for each frequency band (1/3 octave) is determined by means of iso-annoyance curves. The overall annoyance is thus given by a simple formula (analogous to that used in the Stevens method for evaluating overall sonics).

$$N_t = N_{max} + 0.15\ (\Sigma N_i - N_{max})$$

where N_{max} denotes the greatest partial annoyance value;

ΣN_i denotes the sum of all the partial annoyances.

From the total annoyance expressed in ann one may thus compute the perceived noise level using the equation:

$$N_t = 2^{\frac{(PNL) - 40}{10}}$$

where (PNL) denotes the perceived noise level in PNdB and N_t denotes the annoyance in ann. /295

This equation may be approximated by:

$$(PNL) = 33.3 \log N_t + 40.$$

2.2.2. Pure Tone Correction

This method of correction permits definition of a more regular spectrum by two successive smoothing operations, using the characteristics of the measured spectrum. The divergence between the measured spectrum and the corrected spectrum will furnish a correction (termed "C") to be added to the perceived noise level (PNL computed in the first stage).

When this procedure has been completed one obtains the perceived noise level corrected for irregularities in the spectrum, which is abbreviated in English as TPNL.

It can thus be stated that TPNL = PNL + C.

2.2.3. Time Correction

Using the results of the preceding stages it is possible to use points to construct a curve giving the corrected perceived noise level as a function of time. (Here one point was used for each 0.5 second.) This curve (Fig. 1, for example) makes it possible to find:

-- the value for the maximum TPNL, in turn abbreviated TPNLM;

-- the time interval $t_2 - t_1$ by which TPNL exceeds TPNLM: 10 PNdB.

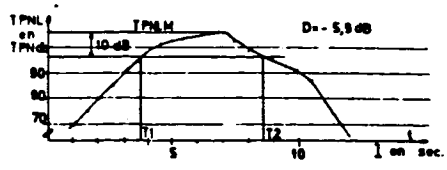

Fig. 1. Key: 1. In seconds.

Determination of $t_2 - t_1$ yields the time correction factor:

$$D = 10 \log \sum_{i=1}^{24} \text{antilog}\left(\frac{TPNL(i)}{10}\right) - TPNLM - 13$$

2.2.4. Effective Noise Level

Finally, the effective perceived noise level, which is the total subjective effect, is given by:

$$EPNL = \overbrace{PNLM + C}^{TNLM} + D$$

III. Existing Measurement Systems /296

Complete systems for measurement of the EPNL have been on the market for some time now (for example, the Brüel & Kjaer system, the Hewlett Packard system, etc.).

The composition of these systems is given in Fig. 2.

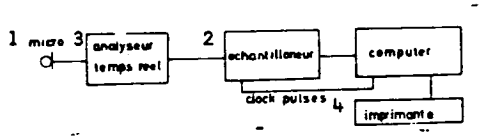

Fig. 2. Key: 1. Microphone. 3. Real time analyzer.
2. Sampler. 4. Printer.

In real time, the analyzer continuously analyzes the sound level in the various bands of frequency. The sampler, commanded every 1/2 second by a signal derived from the clock pulses of the computer, permits storage of the 24 levels characteristic of a spectrum at a given instant.

The computer thus collects the different spectra and converts them, as has previously been shown, into the perceived noise level (PNL); it then computes and adds the C correction and memorizes the TPNL value.

When the test has been completed the computer makes the time correction and transmits the effective noise level EPNL to a teletype.

In addition, the computer program may include a subroutine detecting any important increase in the sound level corresponding to the passage of an aircraft and assuring that the processor is set in operation. The same subroutine will stop the acquisition of data when the level has returned to a sufficiently low point.

It is obvious at first glance that the structure of the measurement system has been dictated by the method. In effect, only a real time analyzer simultaneously monitoring the 24 frequency bands would be fast enough to supply two complete spectra per second.

Furthermore, the use of a computer was made necessary by the large number of data to be processed; for example, a 30-second analysis requires computation of 24 X 30 X 2 = 1,440 partial annoyances in the Kryter tables and the processing of 60 pure tone correction tables.

This type of assembly, however, represents a very large and very specialized investment whose amortization could be assured only by undertaking a large program of research in this area.

IV. The Method of Measurement Developed Using Conventional Equipment

The object of this study was to develop a means of measurement which would use equipment commonly found in any electroacoustics laboratory.

It was also necessary to attempt to:

-- automate the method so as to gain a maximum amount of time

-- simplify the keying in order to use personnel not specially trained for this type of work.

4.1. Acquisition of Data

An initial solution was to record the noise on magnetic tape and to perform the analysis by means of closed loops.

However, the minimum loop length for the equipment which was available to us was such that the time for analysis was 1.7 sec, which was far from one analysis every 1/2 sec. This solution was abandoned in favor of the one shown in diagram form in Fig. 3.

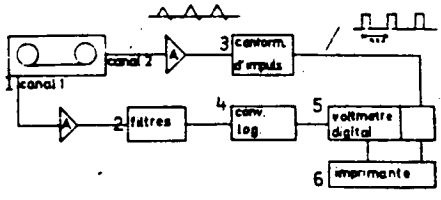

Fig. 3. Key: 1. Channel 1 (typ.). 2. Filters. 3. Pulse shaper. 4. Log converter. 5. Digital voltmeter. 6. Printer.

This solution consists in recording the sound signal (passage of the aircraft) on Channel 1 of a reel of magnetic tape with a second channel containing prerecorded electrical pulses at time intervals of 1/2 sec.

The equipment located at the output of the two channels includes:

a) Channel 1: a Bruel & Kjaer type 2607 amplifier and a type 1614 1/3-octave filter. This assembly is designed to bring the signal to a sufficient level, detect its RMS

value and convert this value into continuous voltage proportional to its logarithm. (In addition this equipment may very earily be replaced by a Brüel & Kjaer type 2111 spectrometer in series with a Bruel & Kjaer type 2305 logarithmic level recorder equipped with "Analog Voltage Read-Out 7R0021.") The output of the log converter is connected to the measurement input of a digital voltmeter.

b) Channel 2: a pulse shaper designed to return the electrical pulses read by the recorder to their original form. (This shaper is actually superfluous, but it increases the reliability of the unit by preventing the transmission of parasitic pulses.) The output of the pulse shaper is linked to the input of a circuit which triggers sampling by the digital voltmeter.

This assembly operates as follows: while the magnetic tape /298
makes its first pass through the unit, the range to be analyzed is defined on the tape by means of two reference marks (adhesive tape). The area thus designated on the magnetic tape is then reproduced 24 times through the 24 1/3-octave filters included between 50 and 10,000 Hz.

During these various passes, the synchronizing signals make it possible for the digital voltmeter to display and the printer to record the values for the sound levels (in dB) within each 1/3-octave band at instants t_1, $t_2 \ldots t_n$, which are always identical and occur at intervals of 1/2 sec.

It is then possible to reconstruct the complete sound spectra every 1/2 sec.

4.2. Processing the Information

4.2.1. The Processing Unit

As has been shown, the number of data involved made it necessary to handle the data automatically by means of a processor. A WANG 700B office minicomputer with the following principal characteristics was used in our electroacoustics department: 120 12-digit stores; 16 possibilities for logical decision; programs capable of including up to 960 steps, storable on magnetic tape.

These characteristics actually define the minimum performances of the processing unit. Obviously a more powerful computer may also be used, on condition that the program diagrams be converted into a language compatible with the system.

V. Conclusion

A method of measurement designed in this way may constitute a valuable basis for the study of aircraft noise by an electroacoustics department. Obviously in addition the data processing programs will permit fast evaluation of the annoyance of a given noise (traffic noise, factory noise, etc.). In this type of measurement, tabulation of the discomfort felt by the hearer is of prime importance. Without the aid of a processor, however, this type of calculation is lengthy and complex, and consequently it has been little used. Thus in our opinion this processing method constitutes a contribution to all problems of noise study.

A detailed bibliography and the process charts for the processing programs are given in the article "Utilisation d'un appareillage classique pour la mesure du niveau de bruit perçu effectif d'un avion" [Use of Conventional Equipment to Measure the Effective Perceived Noise Level of Aircraft] published in the Bulletin Electroacoustique No. 19, July 1971 (Electroacoustics Department, University of Liège).

N74-28400

AIRCRAFT NOISE IN THE REGION OF THE BUCHAREST-OTOPENI AIRPORT

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1. Introduction

The development of air travel throughout the world, which is the result of its advantages over land and sea travel, has led to a remarkably rapid increase in the number of aircraft of different capacities and sizes that are to be found at the same time in the air in different phases of flight. /301

The implications of this phenomenon are extremely complex and not the least problem that requires investigation is that of the noise generated by aircraft at airports and in inhabited regions near them.

In industrialized countries with advanced air transport noise and vibrations are already raising problems of a medical, social, political and economic nature and international cooperation in this field is becoming easier for the purpose of drawing up regulations and directives for the attenuation of the noxious effects of noise and vibrations on man. Thus, 1971 saw the publication of the first edition of international norms, practices and recommencations with regard to aircraft noise -- annex 16 -- by the International Civil Aviation Organization (ICAO), applicable as of 1972.

In Rumania, research on aircraft noise as a source of environmental pollution constitutes a relatively new concern, the present study being the first complex approach to the problem of the noise generated by air traffic in the region of the airports of Rumania.

2. Criteria for Evaluating Aircraft Noise

In view of the great variety of aircraft it is very difficult to find an objective method of measuring their noise. When the problem of aircraft noise is regarded in the light of its effects on human perception it becomes obvious that the data furnished by measuring instruments must be placed into correspondence with those detected by the organ of hearing. The degree of acoustic discomfort generated by an aircraft in flight is determined, according to Ref. [1], by the "Effective Perceived Noise Level" (EPNL), with /302 the measurement unit EPNdB. The effective perceived noise level is given by the instantaneously perceived noise level corrected for spectral irregularities (by means of pure tone correction) and duration (by means of duration correction).

In the event of a succesion of aircraft operations [1] the "Equivalent Continuous Perceived Noise Level" (ECPNL) must be determined with the aid of the following equation:

$$\text{ECPNL} = 10 \log \left[\frac{1}{T} \sum_{i=1}^{n} 10^{(\text{EPNL}_i/10)} \right] \quad \text{(EPNdB)} \tag{1}$$

in which: T is the total reference time in s; n is the number of all operations (landings and take-offs) during T; EPNL_i is the effective perceived noise level for event i.

Studies by sociologists have shown that the effect of noise on man depends on the time of day at which it acts: day (7 AM - 7 PM), evening (7 PM - 10 PM) or night (10 PM - 7 AM). In view of the foregoing it is necessary to determine the "Weight Equivalent Continuous Perceived Noise Level" (WECPNL) in three periods of the day and correct it for the season by means of the following equation:

$$\text{WECPNL} = 10 \log \left[\frac{1}{2} \text{antilog} \frac{\text{ECPNLD}}{10} + \frac{1}{8} \text{antilog} \frac{\text{ECPNLE} + 5}{10} + \frac{3}{8} \text{antilog} \frac{\text{ECPNLN} + 10}{10} \right] + S \quad \text{(EPNdB)} \tag{2}$$

in which: ECPNLD is ECPNL during the day, in EPNdB; ECPNLE is ECPNL during the evening, in EPNdB; ECPNLN is ECPNL during the night, in EPNdB; S is the correction for the season.

In order to bring to light zones of acoustic protection we traced WECPNL = 80 EPNdB and WECPNL = 90 EPNdB curves on the airport map in accordance with the sociological studies of Ref. [1]. In this manner the following zones of acoustic protection are delimited (Fig. 1):

Fig. 1.

-- Zone I: WECPNL > 90, an unpopulated zone where no public buildings can be built. Airport buildings will be perfectly soundproofed. /303

-- Zone II: 80 < WECPNL < 90, which is not recommended for residential quarters. If it is necessary to build residences they must be protected against noise.

-- Zone III: WECPNL < 80, which is the zone in which residential quarters can be built. In the zone WECPNL = 80 the building of hospitals, old peoples' homes, rest homes, schools, nurseries, etc., will be avoided.

3. Aircraft Noise in the Region of the Bucharest-Otopeni Airport and Its Evaluation

In order to determine aircraft noise in the region of the Bucharest-Otopeni Airport we set up our measuring sites in such a manner as to cover the greatest possible land surface in the vicinity of the runway. The topography of the airport permitted adoption of symmetry with respect to the runway. The measurement sites are shown in Fig. 2.

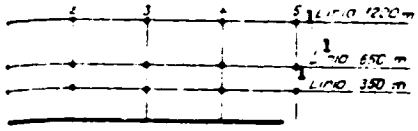

Fig. 2. 1. Line.

The noise level was determined in dBA with the aid of five Brüel & Kjaer 2203 sonometers adjusted for weightedness according to the A curve and a slow dynamic characteristic. We saw to it that our determinations were conducted under meteorological conditions that satisfy the ICAO recommendations concerning temperature, relative humidity, wind velocity, etc., a fact which led to a considerable lengthening of the time alloted to measurements. This length of time was also determined by the size of the land surface investigated, repetition of the determinations for the same measurement site and the same type of aircraft for statistical interpretation of the reduced number of operations.

Using the results obtained in dBA, we determined the values of the effective perceived noise level (EPNL) in EPNdB for take-off and landing operations. Knowing the EPNL values for representative aircraft, i.e., those aircraft that carry out a great number of daily operations, it was possible to plot for each type of representative aircraft equal effective perceived noise level curves (EPNL = const.). These curves are of great utility inasmuch as they make it possible to know the zones with the maximum noise levels for each type of aircraft.

Fig. 3 shows the EPNL = const. curves for take-off operations /304 of BAC 1-11 aircraft.

Knowing the values of the effective perceived noise level EPNL for each type of aircraft and the number of operations for each type of aircraft within a given period of time, it is possible to calculate the equivalent continuous perceived noise level according to Eq. (1). On this basis the weight equivalent continuous perceived noise level WECPNL is determined Eq. (2).

The weight equivalent continuous perceived noise level WECPNL was calculated for a period of seven days inasmuch as analysis of Bucharest-Otopeni Airport traffic for a period of

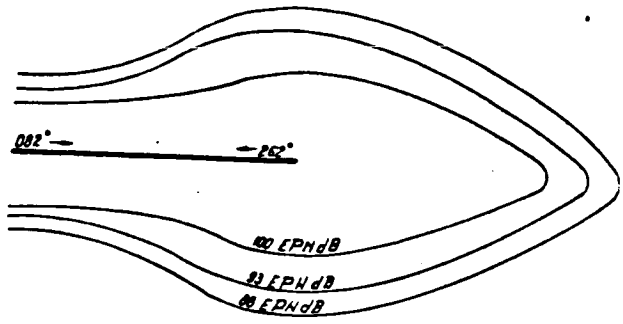

Fig. 3.

a year indicated a very close repetition of the weekly pattern. These considerations permitted selection of the maximum values of WECPNL from the values obtained on weekdays and at every measurement site. With their aid we plotted the curves WECPNL = 80 and WECPNL = 90 (Fig. 4) by which manner the acoustic zoning of the airport region was effected.

Conclusions

a) International cooperation for the purpose of drawing up regulations and directives for attenuating the noxious effects of aircraft noise is being conducted through the International Civil Aviation Organization. The criteria for evaluation of aircraft noise presented in the present study were adopted in conformity with the ICAO recommendations of Annex 16. /305

b) The Bucharest-Otopeni Airport region was zoned as a function of the value of the weight equivalent continuous perceived noise level WECPNL in the following manner:

-- zone I WECPNL > 90 EPNdB;

-- zone II 80 < WECPNL < 90;

-- zone III WECPNL < 80 EPNdB.

c) All inhabited regions in the vicinity of Bucharest-Otopeni Airport are located in acoustic protection zone III.

Fig. 4.

Summary

Aircraft noise, especially in the region adjoining airports, constitutes a problem that will be aggravated in the near future because of increasing aircraft traffic and the appearance of new types of large-tonnage aircraft with continuously increasing powers and speeds.

The present study gives criteria for the evaluation of aircraft noise and some results of studies carried out in the region of Bucharest-Otopeni Airport.

REFERENCES

1. x x x , Aircraft noise in the vicinity of airports ICAO Annex 16. /305

N74-28401

ACOUSTIC WAVE TRANSMISSION THROUGH MULTILAYER MEDIA

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One method of noise control is passive protection by means of which it is intended to increase the resistance of the transmitting medium to acoustic waves [1, 2]. Very often utilized for this purpose are screens formed of many layers of different materials. In the present paper we shall study sound transmission between two media separated by a screen extended to infinity and consisting of three different layers (Fig. 1). /307

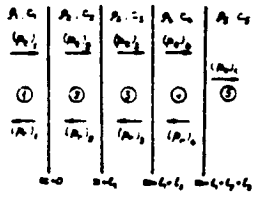

Fig. 1

We shall assume that the two media have characteristic impedances $\rho_1 c_1$ and $\rho_5 c_5$ and that the media that form the screen have characteristic impedances $\rho_2 c_2$, $\rho_3 c_3$, $\rho_4 c_4$ and the thicknesses l_1, l_2 and l_3, where ρ denotes the density of the medium and c denotes the velocity of the sound through the medium.

Acoustic waves are transmitted from medium 1 toward medium 5 by passing through media 2, 3, 4 which form the screen. Owing to the reflection that takes place in the plane of separation of media 1 and 2 ($x = 0$) there will exist in medium 1 an incident wave and a reflected wave:

$$(p_i)_1 = P_{i_1} e^{jk_1(c_1 t - x)} ;$$
$$(p_r)_1 = P_{r_1} e^{jk_1(c_1 t + x)} ;$$

where P_{i_1}, P_{r_1} represent the amplitudes of the incident and reflected waves and k_1 is the wave number.

In medium 2 there will exist a wave transmitted from medium 1 and owing to the reflection that takes place in the plane of separation of media 2 and 3 ($x = l_1$), there will also exist a reflected wave: /308

$$(p_t)_2 = P_{t_2} e^{jk_2(c_2 t - x)} ; \quad (p_r)_2 = P_{r_2} e^{jk_2(c_2 t + x)}$$

In medium 3 there will exist a wave transmitted from medium 2 and owing to the reflection in the plane of separation of meda 3 and 4 ($x = l_1 + l_2$) there will also exist a reflected wave:

$$e^{jk_3[c_3 t - (x - l_1)]} ; \quad (p_r)_3 = P_{r_3} e^{jk_3[c_3 t + (x - l_1)]}$$

In an analogous manner in medium 4 there will exist the following waves:

$$[c_4t-(x-l_1-l_2)]\ ;\ (p_r)_4=P_{r_4}e^{jk_4[c_4t+(x-l_1-l_2)]}$$

In medium 5 there will exist only a single wave transmitted across the plane of separation of media 4 and 5 ($x = l_1 + l_2 + l_3$):

$$(p_t)_5 = P_{t_5}e^{jk_5[c_5t-(x-l_1-l_2-l_3)]}$$

because as the sound is propagated through the five media the wave frequency remains the same:

$$k_1c_1=k_2c_2=k_3c_3=k_4c_4=k_5c_5=\omega = 2\pi f$$

where ω and f denote, respectively, the pulsation and the frequency of the sound.

In order to study sound transmission we shall determine the coefficient of acoustic transmission whose expression in this case has the form:

$$\tau = \frac{P_{t_5}^2}{P_{i_1}^2}\cdot\frac{\rho_1 c_1}{\rho_5 c_5}$$

Knowing the characteristic impedances of the media, we will have to determine the ratio between the acoustic pressure of the wave from medium 5 and the acoustic pressure of the incident wave from medium 1.

For this purpose we write the relations of the condition at the limits of separation of the media.

For x = 0, the continuity of the acoustic pressures and speeds of the particle lead to the relations:

$$P_{i_1}+P_{r_1}=P_{t_2}+P_{r_2}\ ,\quad \rho_2c_2(P_{i_1}-P_{r_1})=\rho_1c_1(P_{t_2}-P_{r_2}) \qquad (1)$$

In an analogous manner we write the relations of the condition at the limits of separation of the other media: $x = l_1$, $x = l_1 + l_2$, $x = l_1 + l_2 + l_3$. Thus we get the relations:

$$P_{t_2}e^{j(k_2l_2)}=\rho_2c_2(P_{t_2}-P_{r_2}) \qquad (2)$$

(3)

(4)

Eliminating P_{r_1} from Eqs. (1) we get: /309

$$P_{i_1} = \frac{1}{2}\left[\left(1 + \frac{\rho_2 c_2}{\rho_1 c_1}\right)P_{t_2} + \left(1 - \frac{\rho_2 c_2}{\rho_1 c_1}\right)P_{r_2}\right] \tag{5}$$

From Eqs. (2) we get:

$$P_{t_2} = \frac{1}{2}\left[\left(1 + \frac{\rho_2 c_2}{\rho_3 c_3}\right)P_{t_3} + \left(1 - \frac{\rho_2 c_2}{\rho_3 c_3}\right)P_{r_3}\right]e^{jk_2 h_2}, \quad P_{r_2} = \frac{1}{2}\left[\left(1 - \frac{\rho_2 c_2}{\rho_3 c_3}\right)P_{t_3} + \left(1 + \frac{\rho_2 c_2}{\rho_3 c_3}\right)P_{r_3}\right]e^{jk_2 h_2} \tag{6}$$

In the same manner, from Eqs. (3) and (4) we get:

$$P_{t_3} = \frac{1}{2}\left[\left(1 + \frac{\rho_3 c_3}{\rho_4 c_4}\right)P_{t_4} + \left(1 - \frac{\rho_3 c_3}{\rho_4 c_4}\right)P_{r_4}\right]e^{jk_3 h_3}, \quad P_{r_3} = \frac{1}{2}\left[\left(1 - \frac{\rho_3 c_3}{\rho_4 c_4}\right)P_{t_4} + \left(1 + \frac{\rho_3 c_3}{\rho_4 c_4}\right)P_{r_4}\right]e^{jk_3 h_3} \tag{7}$$

$$P_{t_4} = \frac{1}{2}\left(1 + \frac{\rho_4 c_4}{\rho_5 c_5}\right)P_{t_5}\, e^{jk_4 h_4}, \quad P_{r_4} = \frac{1}{2}\left(1 - \frac{\rho_4 c_4}{\rho_5 c_5}\right)P_{t_5}\, e^{-jk_4 h_4} \tag{8}$$

Using the system formed by Eqs. (5), (6), (7) and (8) and Euler's relations we get the value of P_{i_1} as a function of P_{t_5} in the form:

$$\begin{aligned} P_{i_1} = \frac{P_{t_5}}{2}\Big[&\left(1 + \frac{\rho_5 c_5}{\rho_1 c_1}\right)\cos k_4 h_4 \cos k_3 h_3 \cos k_2 h_2 + j\left(\frac{\rho_4 c_4}{\rho_1 c_1} + \frac{\rho_5 c_5}{\rho_4 c_4}\right)\sin k_4 h_4 \cos k_3 h_3 \cos k_2 h_2 + \\ &+ j\left(\frac{\rho_3 c_3}{\rho_1 c_1} + \frac{\rho_5 c_5}{\rho_3 c_3}\right)\cos k_4 h_4 \sin k_3 h_3 \cos k_2 h_2 - \left(\frac{\rho_2 c_2}{\rho_1 c_1} + \frac{\rho_5 c_5}{\rho_2 c_2}\right)\sin k_4 h_4 \sin k_3 h_3 \cos k_2 h_2 + \\ &+ j\left(\frac{\rho_2 c_2}{\rho_1 c_1} + \frac{\rho_5 c_5}{\rho_2 c_2}\right)\cos k_4 h_4 \cos k_3 h_3 \sin k_2 h_2 - \left(\frac{\rho_3 c_3}{\rho_1 c_1} + \frac{\rho_5 c_5}{\rho_3 c_3}\right)\sin k_4 h_4 \cos k_3 h_3 \sin k_2 h_2 - \\ &- \left(\frac{\rho_4 c_4}{\rho_1 c_1} + \frac{\rho_5 c_5}{\rho_4 c_4}\right)\cos k_4 h_4 \sin k_3 h_3 \sin k_2 h_2 - j\left(\frac{\rho_5 c_5}{\rho_1 c_1} + \frac{\rho_5 c_5}{\rho_4 c_4}\right)\sin k_4 h_4 \sin k_3 h_3 \sin k_2 h_2\Big] \end{aligned} \tag{16}$$

Writing $\frac{\rho_i c_i}{\rho_j c_j} = k_{ij}$ the relative magnitude of the characteristic impedances of the media we get for the coefficient of transmission the expression:

$$\tau = \frac{4k_{12}(1+k_{23}^2)(1+k_{34}^2)(1+k_{45}^2)}{\left[(1+k_{12})-(k_{23}k_{34}+k_{45})k_{23}k_{34}k_{45}-(k_{23}k_{34}+k_{45})k_{23}k_{34}k_{45}\right]^2 + \left[(k_{23}+k_{34})k_{23}k_{34}+(k_{23}+k_{34})k_{23}k_{34}k_{45}-(k_{23}k_{34}+k_{34}k_{45})k_{23}k_{34}k_{45}\right]^2} \tag{17}$$

It can be seen from this expression that the coefficient of acoustic transmission depends on the relative magnitude of the characteristic impedances of the media through which the acoustic waves are being propagated, the frequency of the sound and the thickness of the layers that constitute propagation media 2, 3 and 4.

Next we shall analyze the tranmission of acoustic waves through the screen considered with due regard for layer thickness. /310

If $k_2 l_1 = n\pi$, where $l_{1n} = n\pi/k_1$ ($n = 0, 1, 2, 3, \ldots$) the coefficient of transmission becomes:

$$\tau = \frac{4\beta_{34}(\alpha+\beta_{45})(\alpha\beta_{45}+\beta_{34})}{[(\alpha+\beta_{45})-(\beta_{34}+\beta_{34}\beta_{45}\alpha)\gamma_3 k_3 l_3]^2+[(\beta_{34}+\beta_{45})\gamma_3 k_3 l_3+(\beta_{45}+\beta_{34}\alpha)\gamma_3 k_3 l_3]^2}$$

In this case the coefficient of acoustic transmission is determined by the characteristic impedances of the media except the second, the frequency of the sound and the thicknesses of media 3 and 4.

If $k_3 l_2 = n\pi$, where $l_{2n} = n\pi/k_3$ ($n = 0, 1, 2, 3, \ldots$), then the expression for the coefficient of transmission becomes:

$$\tau = \frac{4\beta_{34}(\alpha\beta_{45}\gamma_3 k_3 l_3)}{(\alpha+\beta_{34})^2+(\beta_{45}+\beta_{34}\alpha)^2\gamma_3^2 k_3 l_3}$$

It can be observed that the transmission is influenced only by the impedances of media 1, 4, 5, the frequency of the sound and the thickness of medium 4. If at the same time also $k_4 l_3 = n\pi$, i.e., $l_{3n} = n\pi/k_4$ ($n = 0, 1, 2, 3, \ldots$), then:

$$\tau = \frac{4\beta_{45}}{(\alpha+\beta_{45})^2}$$

The expression for τ is analogous to that known from acoustic wave propagation from one medium to another [2].

The intermediate media do not influence sound transmission. If the first and last medium have the same characteristic impedance, then $\tau = 1$ and so the entire energy is transmitted from the first to the last medium.

For $k_2 l_1 = n\pi$, where $l_{1n} = n\pi/k_2$ ($n = 0, 1, 2, 3, \ldots$) and $k_3 l_2 = (2n-1)\pi/2$, where $l_{2n} = (2n-1)\pi/2k_3$ ($n = 0, 1, 2, 3, \ldots$), we get:

$$\tau = \frac{4\beta_{34}(\alpha+\gamma_3^2 k_3 l_3)}{(\beta_{34}+\beta_{45}\alpha)^2\gamma_3^2 k_3 l_3+(\beta_{34}+\beta_{45})^2}$$

Transmission is influenced by the characteristic impedances of the media except the second, the frequency of the sound and the thickness of the fourth medium. If at the same time $k_4 l_3 = n\pi$, where $l_{3n} = n\pi/k_4$ ($n = 0, 1, 2, 3, \ldots$) then:

$$\tau = \frac{4\beta_{45}}{(\beta_{45}+\beta_{34})^2}$$

Only the first, third and fifth medium influences sound transmission.

In order to have integral transmission of acoustic energy from the first to the last medium it is necessary to have $\tau = 1$, which leads to:

$$\rho_3 c_3 = \rho_4 c_4 \cdot \rho_5 c_5$$

Therefore, the characteristic impedance of the third medium must be the geometric mean of the characteristic impedances of the first and fifth medium. /311

If the first and last medium are identical or have the same characteristic impedance we shall have integral transmission of the energy in $\rho_1 c_1 = \rho_3 c_3 = \rho_5 c_5$.

If, however, $k_4 l_3 = (2n-1)\pi/2$, where $l_{3n} = (2n-1)\pi/2k_4$ $(n = 1, 2, 3, \ldots)$ then:

$$\tau = \frac{4\rho_1 c_1}{(\rho_3 c_3 + \rho_5 c_5)^2}$$

The coefficient of transmission depends on the characteristic impedances of the media except the second.

In order to have integral transmission we get the condition:

$$\left(\frac{\rho_3 c_3}{\rho_4 c_4}\right)^2 = \frac{\rho_1 c_1}{\rho_5 c_5}$$

Therefore, the square of the relative characteristic impedance of the third medium referred to the fourth medium must be equal to the relative characteristic impedance of the first medium referred to the fifth medium.

A special case that may occur is when:

$$k_2 l_2 = \frac{(2n-1)\pi}{2}, \quad l_{2n} = \frac{(2n-1)\pi}{2k_2}; \quad k_3 l_3 = \frac{(2n-1)\pi}{2}, \quad l_{3n} = \frac{(2n-1)\pi}{2k_3}, \quad k_4 l_4 = \frac{(2n-1)\pi}{2},$$
$$l_{4n} = \frac{(2n-1)\pi}{2k_4} \quad (n = 1, 2, 3, 4, \ldots)$$

The coefficient of transmission is:

$$\tau = \frac{4\rho_1 c_1}{(\rho_3 c_3 + \rho_5 c_5)^2}$$

and depends only on the characteristic impedances of the media.

In order to have total transmission we get the condition:

$$\left(\frac{\rho_3 c_3}{\rho_4 c_4}\right)^2 = \frac{\rho_1 c_1 \cdot \rho_5 c_5}{(\rho_4 c_4)^2}$$

Therefore, integral transmission of energy from the first to the last medium takes place when the square of the relative characteristic impedance of the second medium referred to the third medium is equal to the ratio between the product of the characteristic impedances of the first and last medium and the square of the characteristic impedance of the fourth medium.

In Table 1 we have summarized the situations that can occur depending on the thicknesses of the three intermediate layers.

TABLE 1

Layer thickness			s	Integral energy transmission condition
l_{1n}	l_{2n}	l_{3n}		
$\frac{n\pi}{k_1}$	$\frac{n\pi}{k_2}$	$\frac{n\pi}{k_3}$	$\frac{4\beta_1\beta_3}{(\beta_1+\beta_3)^2}$	$\rho_1 c_1 = \rho_3 c_3$
$\frac{n\pi}{k_1}$	$\frac{(2n-1)\pi}{2k_2}$	$\frac{n\pi}{k_3}$	$\frac{4\beta_2\beta_3}{(\beta_2+\beta_3)^2}$	$\rho_2^2 c_2^2 = \rho_1 c_1 \rho_3 c_3$
$\frac{n\pi}{k_1}$	$\frac{(2n-1)\pi}{2k_2}$	$\frac{(2n-1)\pi}{2k_3}$	$\frac{4\beta_2\beta_3}{(\beta_2+\beta_3)^2}$	$\left(\frac{\rho_2 c_2}{\rho_3 c_3}\right)^2 = \frac{\rho_1 c_1}{\rho_3 c_3}$
$\frac{n\pi}{k_1}$	$\frac{n\pi}{k_2}$	$\frac{(2n-1)\pi}{2k_3}$	$\frac{4\beta_3\beta_2}{(\beta_3+\beta_2)^2}$	$\rho_3^2 c_3^2 = \rho_1 c_1 \rho_2 c_2$
$\frac{(2n-1)\pi}{2k_1}$	$\frac{n\pi}{k_2}$	$\frac{n\pi}{k_3}$	$\frac{4\beta_1\beta_2}{(\beta_1+\beta_2)^2}$	$\rho_1^2 c_1^2 = \rho_2 c_2 \rho_3 c_3$
$\frac{(2n-1)\pi}{2k_1}$	$\frac{(2n-1)\pi}{2k_2}$	$\frac{(2n-1)\pi}{2k_3}$	$\frac{4\beta_1\beta_2\beta_3}{(\beta_1\beta_2+\beta_2\beta_3+\beta_3\beta_1)^2}$	$\left(\frac{\rho_1 c_1}{\rho_2 c_2}\right)^2 = \frac{\rho_3 c_3}{\rho_2 c_2}$
$\frac{(2n-1)\pi}{2k_1}$	$\frac{(2n-1)\pi}{2k_2}$	$\frac{n\pi}{k_3}$	$\frac{4\beta_1\beta_2}{(\beta_1+\beta_2)^2}$	$\left(\frac{\rho_1 c_1}{\rho_2 c_2}\right)^2 = \frac{\rho_3 c_3}{\rho_2 c_2}$
$\frac{(2n-1)\pi}{2k_1}$	$\frac{n\pi}{k_2}$	$\frac{(2n-1)\pi}{2k_3}$	$\frac{4\beta_1\beta_3}{(\beta_1+\beta_3)^2}$	$\left(\frac{\rho_1 c_1}{\rho_3 c_3}\right)^2 = \frac{\rho_2 c_2}{\rho_3 c_3}$

In practice, cases occur when it is intended for the sound energy to be transmitted integrally, as is the case of acoustic waves from water into the spongy rubber employed in protecting the radiating surface of electroacoustic transducers that work under water, or for the integral transmission of the energy to be impeded. /312

Taking account of the expression $k = \frac{\omega}{c} = \frac{2\pi f}{c}$ the relations between layer thickness and sound frequency can be established.

For $l_n = n\pi/k$ we get $l_n = \frac{nc}{2f}$ and for $l_n = \frac{(2n-1)\pi}{2k}$ we get $l_n = \frac{(2n-1)c}{4f}$.

If the sound frequency that must be transmitted is known and the relations between the characteristic acoustic impedances of the media for integral transmission of energy are satisfied, then taking into account the nature of the layer their thickness can be determined in order to bring aboug integral energy transmission or the thickness can be chosen in such a manner that this transmission is impeded. More difficult to bring about is total transmission, for it is also influenced by other factors, among which internal friction plays an important role.

Summary

The authors study acoustic wave transmission through several media extended to infinity. They analyze the factors that influence transmission. They determine the conditions under which acoustic energy is transmitted integrally. /313

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NOISE AND VIBRATION LEVEL REDUCTION BY COVERING METAL STRUCTURES WITH LAYERS OF DAMPING MATERIALS

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The noise and vibration level of different types of structures /315
can be reduced by a modern, particularly efficient method: covering the structures with layers of materials having the property of damping deformation energy. As is well-known, the classical methods employed for this purpose usually consist in stiffening the elements of the structure, decoupling resonant systems, detuning coupled resonators, constructing structures with materials having appreciable fatigue strength or insulating the source. Unfortunately, in certain cases these solutions are unsatisfactory, especially when a certain structure must operate under very complex dynamic conditions, at temperatures and under stresses that vary within a very wide range of values. Thus, especially in the automobile and rolling stock manufacturing industries, the damping treatment method constitutes a solution that is acceptable from the engineering point of view as well as that of its technological and economical efficiency.

In order to estimate the qualities of treatment by covering with damping materials called mastics, it is important that, on the basis of experiments conducted on simple structures (beams, plates), we should be able to make predictions about the damping qualities of more complicated structures [2, 5, 7, 11]. Inasmuch as the materials employed for this purpose must also possess mechanical strength to different types of static and dynamic stresses, adherence, and anticorrosive properties, noninflammability, mechanical properties that do not vary in time with temperature and stress frequency except within acceptable limits, a small weight /316
without the necessity of a complicated technology for application to structures, as well as a low cost price, it is obvious that total estimation of a certain type of mastic cannot be reduced merely to determination of noise and vibration damping capacity.

Experimentally it has been found [10] that there exists a whole class of materials, including mastics, whose mechanical behavior can be described by linear viscoelastic models. At relatively high temperatures and/or extremely low stress frequencies, they show elastic properties like those of rubber, their capacity to dissipate deformation energy in the form of thermal energy being extremely small. At lower temperatuers and/or higher frequencies these materials become more damping. Finally, at even lower temperatures, below a certain well-defined temperature, and/or at

very high frequencies, these materials become brittle and the energy dissipated is extremely low. The region in which the dissipated energy attains a maximum for a range of temperature and frequency values that vary with the material, is of great interest because of special mechanical properties. Quantitatively, this type of mechanical behavior is completely characterized by the so-called complex modulus of elasticity $E^* = E(1 + i\beta)$, where E is the material's modulus of elasticity and β is the loss (damping) coefficient and expresses the material's vibration damping capacity. Inasmuch as our purpose is not only to use materials with appropriate loss coefficients but also to obtain a high loss coefficient ψ for the structure, it is obvious that the structure's geometry will also play an important role in estimating its damping capacity. The vibrations to which the structures in question are subjected, as well as the noises they transmit, arise through two distinct channels: gripping (supporting) elements and the ambiant medium (air). It should be mentioned that reduction of the level of vibrations as well as that of transmitted noise depends on the internal (volume) dissipation due to friction between the structure's particles, on superficial dissipation (absorption) due to friction between the structure's elements and /317
the ambiant medium and on dissipation due to attenuation through reflection. The extent differs to which each of the above-mentioned mechanisms affects the level of vibrations and of noise transmitted by the structure under consideration. Thus, while the properties of internal dissipation are particularly important in reducing the vibration level, the properties of superficial absorption and of attenuation through reflection strongly influence noise level reduction. Inasmuch as application of damping layers to the surface of the structure's elements generally changes all the properties of the resultant structure and the main purposes of this treatment -- reduction of the structure's vibration level and reduction of the transmitted noise level -- are differently affected by the change in the resultant structure's properties, our study was oriented in two different directions: determination of the efficiency of the treatment on vibration damping on the one hand and on attenuation of the noise transmitted by such a structure, on the other.

Determination of the Efficiency of Covering with Layers of Visco-Elastic Materials for the Purpose of Vibration Damping

Theoretical analysis has been carried out by Oberst [1], Schwarzl [3], Markus [8] and others, who established the expression for the loss factor of beams. Usually the relations obtained are relatively simple but must be satisfied by certain conditions such as: the maximum flexure must be small enough, the loss factor of the compound structure must not be too great (preferably <0.2) and the loss factor of the material of the uncovered structure must be negligible in comparison with that of the

covering material. Schwarzl's analysis, however, is broader in that the last two restrictions are not employed as working hypotheses. It should be stressed [4] that the equation of the Bernoulli-Euler type that lies at the base of the theory of the flexure of compound beams does not take into account effects due to transversal shearing nor rotation inertia which, in the case of thick beams (the ratio length (L)/thickness (H) of the small beam) and at high frequencies of the exciting force, become important. /318
The effect of rotation inertia consists in a drop in the value of the beam's natural frequencies as a function of the values of the ratio L/H (for values of L/H > 100 this effect becomes negligible, but when L/H < 50, the errors due to neglecting it are appreciable [4]). The abnormal results of Preise and Skinner [9] confirm this. Finally, the possibility of neglecting the energy dissipated in the metal structure depends on the ratio of the flexure strength of the compound beam and the uncovered beam (for values of this ratio < 1.1 the effects are very important and neglecting them entails large errors in estimating the damping coefficient of the covering material). In addition to these theoretical limitations, there occur errors due to the experimental method itself.

The experimental method we used is similar to that of Oberst. The metallic beam, covered on one side with a viscoelastic material, is fixed at one end and free at the other. The beam is subjected to a periodic stress of fixed amplitude (generated by an electromechanical transducer at desired frequencies, supplied by a precision oscillator with adjustable frequency) at the free end. The beam's response is recorded through the agency of an electromechanical transducer by an electromechanical millivoltmeter (Fig. 1).

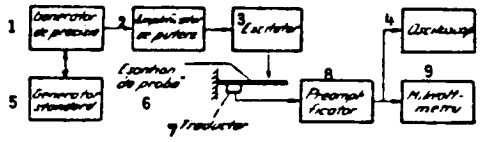

Fig. 1. 1. Precision generator; 2. Power amplifier; 3. Excitor; 4. Oscilloscope; 5. Standard generator; 6. Test piece; 7. Transducer; 8. Preamplifier; 9. Millivoltmeter.

The loss factor of the compound beam is determined at a resonance frequency f_0 chosen as a function of real necessities (200 Hz in our case) from the width Δf of the resonance curve, for an attenuation of 3 dB, using the relation $\psi = \Delta f/f_0$. From the results obtained in this manner the loss factor of the covering material can also be determined, taking into account the geometry of the compound beam and the mechanical properties of the uncovered beam.

Inasmuch as the study [4] of uncovered beams shows that a pre- /319
cision of up to ±0.5% can be attained in determining resonance frequency, this is the lowest error limit in the case of compound beams too; as a matter of fact, depending on the loss factor of the compound beam, the error can even reach ±3%, if the loss

factor is approximately 0.2%. In carrying out our experiments we took special precautions for correct determination of the thickness of the mastic layer, which is of importance to the damping capacity of the compound structure.

Fig. 2 shows the experimental results obtained for the loss factor ψ of a 0.7 mm thick metal (steel) beam covered with seven types of mastics R1, R2, R3, R4, R5, R6 and T, having different thicknesses ranging from 0.4 mm to 2.6 mm. The damping properties of these structures were studied at a frequency of 200 Hz and a temperature of 21° C. As can be seen from Fig. 2, the loss factors of the compound structure decrease in the order R2, R3, R4, R5, R6, T, R1, and mastics T and R1 coincide in the thickness range recommended for mastic T on utilizing it as covering material in the automobile industry. The curves permit the values of the damping coefficient of the compound structure to be obtained for a desired thickness of the covering layer or the thicknesses of the different mastic layers employed as covering material so that the compound beam will have the same loss (damping) coefficient.

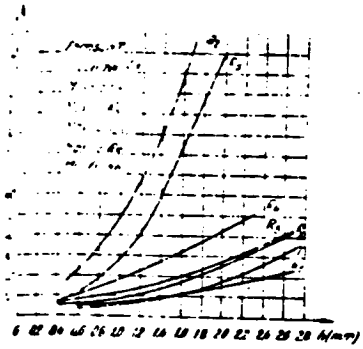

Fig. 2.

Determination of the Attenuation of Noise Transmitted Through Structures Covered with Damping Layers

The transmission of noise through structures can be conveniently characterized on the basis of energy criteria [13], defining the coefficient of transmission τ as the ratio of the acoustic energy flux Φ_t transmitted through the structure studied and the incident flux Φ_i. As a result, the quantity frequently employed to characterize the attenuation of sound transmission through structures will be given by the inverse of τ expressed in decibels, which is called the damping transmission coefficient (or index) D. Inasmuch as the transmitted and incident fluxes are precisely the acoustic intensities I_1 and I_2 at the separation surfaces of the structure, the most appropriate quantity for determining the efficiency of the damping layers for covering a structure is, in fact, the variation in the damping transmission coefficient, i.e.: /320

$$D = D_a - D_n = 10 \lg \frac{I_1}{I_2} - 10 \lg \frac{I_1'}{I_2'} = 10 \lg \frac{I_1'}{I_2} = 20 \lg \frac{p_{1ef}'}{p_{2ef}},$$

in which the quantities D_n, I_2, p_{2ef} refer to the uncovered structure and D_a, I_1', p_{2ef}' refer to the covered structure for the same values of I_i and effective pressure p_{1ef} on the incident surface.

At the present time, the values of the damping transmission coefficient are calculated with a quite unsatisfactory precision because of the different simplifying hypotheses necessary for complete resolution of such a problem. This calculation, which is carried out on the basis of the relation known in acoustics as the law of mass [13], as well as the more exact approximation to the value of D by attempting to take into consideration not only the vibrations of the structure as a rigid piston but also its flexure vibrations, which have led to the working out of the theory of coincidence [14], in certain cases give rise to errors that are much too great as compared with measured values. As a result, existing calculations are utilized even in the case of simple structures only for tentative purposes while for more exact quantitative determination, experimental methods are adopted.

The experimental method we utilized for determining the damping transmission coefficient in the case of structures covered with damping layers is based on the block diagram of Fig. 3. It consists of a setup with absorbing walls into which are introduced simple structures and structures covered with damping layers that form in succession a screen in the sound transmission chain between the diffuser system and the microphone. /321

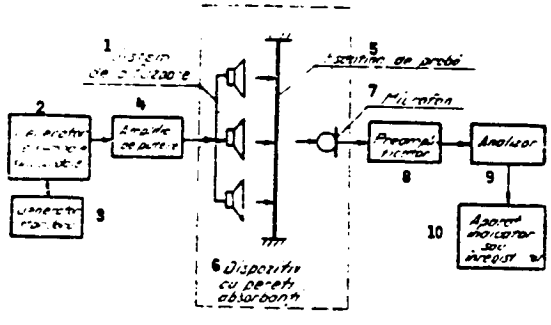

Fig. 3. 1. Diffuser system; 2. Sinusoidal signal generator; 3. Standard generator; 4. Power amplifier; 5. Test piece; 6. 6. Setup with absorbing walls; 7. Microphone; 8. Preamplifier; 9. Analyzer; 10. Indicating or recording instrument.

In building the setup with absorbing walls and choosing the afferent apparatus we took a number of measures to ensure good accuracy of the experimental results: obtaining the most uniform acoustic pressure possible on the incident face of the structure studied, avoiding any reflection in the region of the outer face, etc. Fig. 4 shows the experimental results obtained in the case of a 1 m x 1 m and 0.8 mm thick steel plate fixed on all sides. In the figure the broken line represents the curve that expresses the variation with frequency of the damping transmission coefficient for the uncovered structure, and the solid line represents the same variation in the case of the same structures covered with a 1.2 mm thick mastic layer. It can be seen that the variation with frequency of the difference $D = D_n - D_a$ is relatively small, so that it can be expressed by a mean value $D \simeq 12$ dB in the frequency range studied. /322

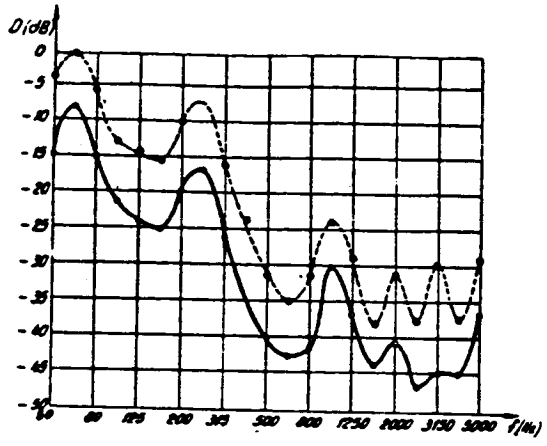

Fig. 4.

Conclusions

In what follows we shall briefly discuss some of the more important conclusions that emerge from our determinations. Determinations of the loss factor by the resonance method are more correct than those based on the damping curves of the free vibrations of beam flexure. The values obtained for beams by the resonance curve method leads to smaller values than those obtained by measuring large plates in the case of uncovered structures; for covered structures the situation is inverted. It is obvious that in addition to the energy damped in the structure, what is automatically being measured in such determinations is the radiated energy. In the case of uncovered bars the radiated energy is low while for plates it becomes greater than the damped energy. In the case of covered structures we can assume [6] that losses through radiated energy drop in comparison with damped energy. Since even in the case of beams it is doubtful that only flexure vibrations are excited, all the more so in the case of plates can different types of vibrations occur and the /323
energy be distributed among different types of vibrations; and, since damping of the energy of flexure vibrations is the most important, it is clear that in the case of covered plates the dissipated energy in the structure will drop accordingly. During the experiments we took measures to reproduce as correctly as possible the conditions of excitation and manners of gripping encountered in practice. The degree of precision of these models must be established more exactly by correlating future experimental results with some theoretical considerations like those of Heckl [15].

Summary

One of the most important methods of reducing the noise and /324
vibration level is the damping of the secondary sources, such as metal plates, often used in vehicle structures, by means of covering materials with high internal viscosity. The effectiveness of this method depends not only on the skillful choosing of the viscoelastic damping layers corresponding to the frequency and temperature range in which a certain structure will work, but also on the choice of an optimum thickness of the covering material. The experimental

method of investigation was carried out by using an electromagnetic exciter which gives the desired frequency range by means of a precision generator with adjustable frequency possibilities. The structure's response corresponding to various real situations, is analyzed by means of a measuring chain including electroacoustical or electromechanical transducers. The experimental results provide the dependence of the loss factor and damping transmission coefficient as a function of the damping layer thickness or of the frequency for various viscoelastic covering materials.

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DIFFERENTIAL PHONOGRAMS

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Problem

Experienced engineers can ferret out defects or derangements /325
by simply listening to the noise radiated by the equipment. But these information-carrying components that are empirically detected by the mature human ear cannot be brought to light by any of the usual methods of acoustic analysis. What is required is something new, something that does not stem from the procedures of noise control.

Solution

We worked out a method that we are provisionally calling the "differential phonogram method". This method models electronically some of the processes that take place in the human organ of hearing and central nervous system.

We sought to eliminate the nonrelevant components that mask the useful information and hamper extraction from the complex of noise radiated by the equipment, of those discrete components that invariably accompany certain running cycles.

On oscilloscope screens or recording tapes the differential phonograms reproduce discrete elements that are characteristic of a certain part or subassembly.

The differential phonograms visualized on the screen or recording paper permit clear recognition of deviations, even by personnel with little experience. Clearly to be recognized are the following:

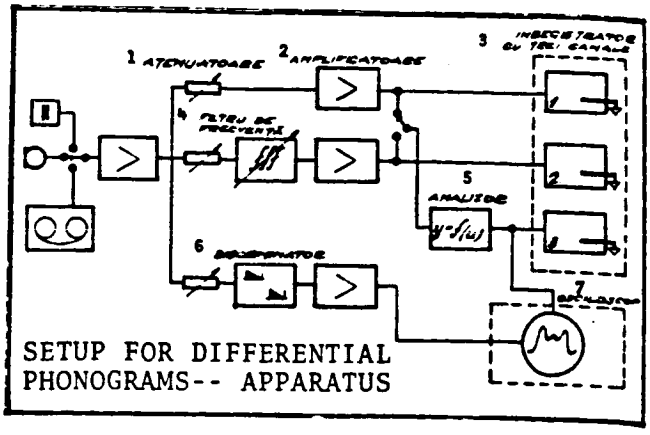
-- deviations from normal running rhythm; /326

-- deviations from normal duration of some phase of running;

-- dissymetry of similar phases;

-- fluctuations in intensity;

-- deviations from normal speed.



SETUP FOR DIFFERENTIAL PHONOGRAMS-- APPARATUS

Fig. 1. 1. Attenuators; 2. Amplifiers; 3. Three-channel recorder; 4. Frequency filter; 5. Analyzer; 6. Discriminator; 7. Oscilloscope.

Experimentation

The block diagram of the setup we used in our experiments is shown in Fig. 1. The noise picked up by microphone and vibration transducer or recorded beforehand on magnetic tape passes through four signal processing lines:

Line 1 consists of a linear amplifier that passes the signal toward one of the recording channels;

Line 2 possesses, in addition, an adjustable frequency filter; on the second channel of the recorder the noise curve /327 of the corresponding frequency band is obtained;

Line 3 consists of the electronic analyzer which can be switched either to the total noise or to the noise in one frequency band; the electronic analyzer works on the third channel of the recorder and on the amplifier Y of an oscilloscope;

Line 4 consists of a discriminator which supplies synchronization pulses for the oscilloscope (for example, for every uneven pick of a loom or every fourth explosion of a four-cylinder motor).

The setup was tested on a loom manufactured by a firm with ancient traditions in the matter. The results provided by the differential phonogram method led to the redesigning of some loom subassemblies (the transmission, the cam mechanism, the guards), permitting the cloth to be made uniform and some sources of premature wear to be eliminated.

Differential Phonograms

Figure 2 shows the recording of 12 picks for five seconds of /328 the loom's running. We used the setup described in the preceding section. Recordings were taken on a Philips oscilloscript. In the phonogram (curve b) the picking moments can be easily recognized, as well as the arhythmicity of picking and the nonuniform duration of shuttle braking. Curve a marks the time in seconds. Curves c and d represent the recording of total noise and that of filtered noise.

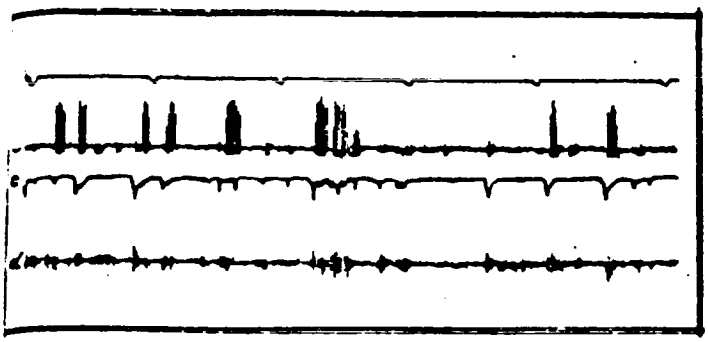


Fig. 2.

The phonogram of Fig. 3 shows the irregularities in the running of the cam mechanism in the same loom.

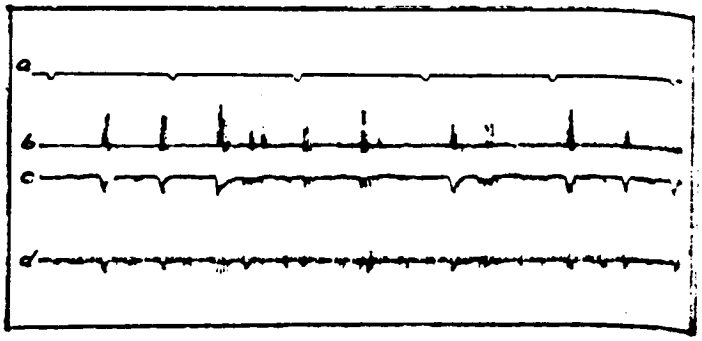


Fig. 3.

It can be seen that at some moments detachments of the roller from the cam profile take place (at the first three consecutive picks). At the fourth pick a repeated detachment takes place, leading also to an imprecise pick of the sword. There follow three picks for which no detachment takes place, then another two detachments.

Examining the phonograms plotted during many minutes of running, it is possible to detect some construction defects. Visualization of the phonogram during adjustment operations permits them to be carried out at extremely close tolerances. Periodic checkup of the equipment by recording the noise on a portable tape recorder and analyzing it in the sound laboratory permit early

detection of wear and derangement in looms that are being manufactured. In this manner we can prevent more serious defects and eliminate some defects in the cloth caused by defects that are difficult to detect by other methods. /329

Summary

Conventional noise measurement and analysis do not offer data on the working state of equipment. /330

A new method is outlined in this paper, based on electronic analysis of the radiated noise. Non-revelant components are suppressed, while the differential phonogram displayed on the oscilloscope screen discloses the information-carrying components. It allows low-skilled operators to perform: (1) correct alignment of equipment; (2) detection of incipient failure and (3) checking of protypes while running in.

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PRESENT TENDENCIES IN EQUIPMENT NOISE NORMALIZATION

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One of the characteristics of modern industry is an increase /331
in the noise level, which has become a noxious factor for the person engaged in the production process. The principal source that sets up this noxious climate is industrial equipment. To this must be added the construction of large industrial plants, the utilization of certain technologies, etc.

Present technology has dictated an increase in the power and productivity of equipment and, at the same time, a reduction in the size and weight of machines with a view to obtaining an increase in their running speeds and, therefore, in dynmaic forces, which have had as a consequence an increase in vibrations and in radiated acoustic power.

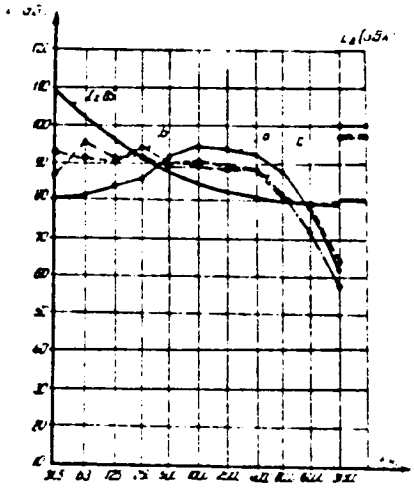
Only in the last 10-20 years has the noxious effect of noise on man come to light. In spite of this, at the present time, in most branches of machine building there still exists no concern for reducing the noise level of equipment.

One of the criteria that are widely recognized at present for estimating the noxiousness of noise for man, set forth in recommendation ISO/R 1999-71, is the deterioration of hearing (a loss of hearing with respect to conversation by a shift in the auditory threshold). In that document noise exposure is linked to hearing loss by means of the concept of risk, which is defined as the expression of the probability that personnel exposed to noise will develop a certain degree of impaired hearing. The noise level is expressed by the weighted acoustic level A, L_A. The acoustic level above which the risk of a loss in hearing of conversation occurs is 80 dBA; the risk increases with the acoustic level and with the duration of noise exposure (expressed in work years).

In our country the limiting values admissible for noise in a work space are established by the Republican Labor Protection Norms (RLPN) by Cz noise curves (the ISO notation is N). These are curves of equal acoustic intensity level. The limiting curves are chosen beginning with the Cz85 curve, which is the most tolerant, /332
as the function of the nature of the work and the degree of attention that is required for the jobs executed in different work spaces. For work spaces situated in shops and industrial plants the admissible limiting curve CZ85 is taken into account. In view of the fact that we are considering industrial equipment and work

spaces situated near them, and in order to situate ourselves in a range recommended by ISO/R 1999-71, which brings to light the accute aspect of the problem -- impaired hearing -- we shall consider that the noxious values are found above the Cz85 noise curve and the acoustic level of 80 dBA.

In order to illustrate to what an extent workers in industry work under inadequate conditions as regards noise exposure, we present a figure showing the noise level in frequency bands and the weighted acoustic level recorded in three industrial plants with different profiles: curve a represents the noise level recorded in the shielded shed of a wool weaving mill in which classical looms with a shuttle run; curve b represents the turbogenerator noise recorded in the machine room of a thermoelectric power plant; and curve c represents the noise level existing in a compression plant. The acoustic pressure level in frequency bands in the three cases is situated above the Cz85 curve, with values that attain 6...14 dB, and the weighted acoustic level A exceeds the value of 80 dBA by 17...20 dBA.



In the practice of noise control /333
it is generally recognized that the most direct and efficient method is that of noise control measures at the source. Reducing the noise level of equipment that is already running in industrial plants is a difficult and sometimes impossible task. This confronts the factory with the problem of acquiring silent equipment from the manufacturing plant. This, however, can only be done on the basis of restrictive norms concerning the admissible level of noise for the equipment in question. The working out of restrictive norms for the level of noise generated by equipment thus becomes a necessary condition for placing the noise problem within the framework of the design and manufacturing tasks of manufacturing plants.

The norms referring to equipment noise have two aspects:

a. Norms that establish the methodology to be adopted in taking acoustic measures with regard to the equipment;

b. Norms for limiting levels for noise generated by the equipment.

The method norms generally have a greater utility. They can serve the following purposes:

-- compare the noise emitted by the equipment built according to the same specifications;

-- compare the noise emitted by equipment of different types;

-- determine the noise radiated for a given distance;

-- check whether the noise generated by a piece of equipment satisfies a certain norm.

The measurement method and the acoustic magnitudes that are determined generally depend on the purpose of the measurement because the use of one or another method leads to more or less exact results for the magnitude that are finally of interest. In the present case the drawing up of norms for methods is of interest from the standpoint of their utilization in working out norms of limiting levels (the study of the noise generated by a piece of equipment) and, subsequently, in checking whether the noise generated by equipment of the same kind satisfies this norm. It is important that the values in which the acoustic measurement results are expressed and the limiting values established by the norm be the same as those for the sanitary norms that estimate the noxious level of the noise.

At the present time it is recognized throughout the world that the values that can yield relations for the noxious effect of noise on man are the weighted acoustic level A and the acoustic pressure level in frequency bands. All standards that deal with equipment noise measurement provide determination of these values by the method of the close acoustic field, or the method of the free acoustic field over a reflecting plane. These methods yield suitable results for estimation of the noxiousness of noise in work spaces situated near the equipment, but have the disadvantage of yielding approximative values in the case of calculating the acoustic power radiated by the equipment. /334

One of the important data that are established within the framework of the methodology to be used in acoustic measurements is the machine's operating conditions. During noise determinations the machine may be running idly or under a full load. If we consider noise measurements in the light of their utilization for establishing useful norms of limiting levels for noise reduction incident to the running of equipment in industrial plants, it is clear that the measurements would have to be carried out incident to the running under full load or the nominal parameters that constitute the real situation. Not lacking in interest either are the measurements conducted when the machine is idling, for they can give

indications regarding the influence that the load has on the noise it generates. General standards referring to the drawing up of research codes for machine noise measurement (national as well as international) leave the specification of operating conditions up to specific standards. The specific standards for noise measurement in electric machines, motor-compressors, pneumatic tools, Diesel engines, fans, etc., established by documents ISO (R1680-1970, R2151-1972), France (S. 30-006), USA (S5.1-1971), Federal German Republic (DIN45635), Socialist Republic of Czechoslovakia (CSN350000, CSN123062, CSN090862), etc., establish that noise measurement be conducted under load at nominal running parameters and sometimes also during idle running. Standardization recommendation CAER/RS675-66, which refers to determination of the noise level in electric machines, is relatively old as compared with the above-mentioned standards, and establishes that acoustic measurements be conducted during idle running and that the noise under load be measured only when it is necessary to know the influence of the load. Measurements during idle running are also provided for by the Rumanian standard in force, STAS 7301-65, which refers to noise measurement in electric machines.

In some cases it is difficult to conduct measurements under load because of the coming into play of a great number of parameters in defining a given set of operating conditions, the great number of materials that may be processed, etc., which makes it difficult for measurement conditions to be reproduced. This problem was solved by the American (corporation) norm S5.1 1971, which refers to the conducting of noise measurements in pneumatic equipment during running under load, with detailed specifications and graphics concerning measurement conditions. This problem is treated in the same manner in the draft of the ISO standard that refers to pneumatic tools and machines. /335 The above-mentioned difficulties have caused the Rumanian standard referring to methods of measuring the noise level in cutting machines, which are about to be published, to establish that noise measurements shall be conducted during the machine's idle running.

Norms referring to the limiting admissible noise level of equipment can have two objectives:

1. They check whether the equipment's design ensures safe operating conditions.

2. They check whether the equipment in question generates a noise level that respects the limits considered noxious for the man who operates it.

At the present time there exist norms of limiting levels that satisfy only the first objective, and these refer especially to rotating electric machines.

Existing norms referring to limiting noise levels for checking safe operating conditions cannot be utilized from the standpoint of checking the noxious effect of the equipment's noise on man, either because of the very tolerant limits, or because of the operating conditions established for the machines that are being tested (idle running: CEI34-9/1972, GOST 16372-70, VDE 0530-1172, STAS 8274-68), or sometimes because of the values in which the limiting noise level is expressed (acoustic power level: CAER/RS 1194).

Norm CEI34/1972, which was recently revised, stipulates for the majority of electric machines higher admissible limiting power and acoustic pressure values than those established by the previous norm. Thus, in electric machines with powers of 22 Kw < P < 400 Kw and more than 1,500 rpm, the admissible limiting values for idle running at a distance of 1 m from the machine are considered to be acoustic levels ranging from 85 to 102 dBA. It follows from the foregoing that the standards of limiting level, which are aimed exclusively at ensuring safe operating conditions, can become the basis of legislation for the manufacture of equipment that generates a noxious noise level.

The problem of working out for equipment standards of limiting noise levels that will prevent generation of a noxious level in the work space is a complex one that requires many stages for its solution, namely:

1. Adoption of norms that will establish criteria and limiting levels for the noxiousness of noise in the work space.

2. Working out of standards concerning the methodology of noise measurement in equipment that will take into account their real operating conditions and will make use of methods and magnitudes that are compatible with those established by the norms that limit the noise level in the work space. /336

3. Systematic acoustic measurements of banks of machines in order to determine their noise level. At the same time it is necessary to study the nature, causes and possibilities of reduction of the noise level of these machines. Work could be begun in research institutes close to the manufacturing plants, with the assistance of the latter or their specialized laboratories.

4. Working out of standards of admissible limiting levels for equipment that will be based on the results of the study of noise and the possibilities of reducing it in the equipment in question (the existence of silent prototypes).

The working out of the standard dealing with the methodology of noise measurement in certain equipment must precede by at least two years the standard of limiting levels, the length of the delay

depending on the difficulty and the stage that the study of noise in the equipment in question has reached.

So far, in Rumania the problem of equipment noise has been tackled by the standards for electric machines dealing with measurement methods (STAS 7301-65) and admissible noise levels (STAS 8274-68). This must be followed in the next few years by revision of these two standards, the appearance of a standard for measuring methods for machine-tools, the working out of standards for measurement methods and admissible noise levels for turbogenerators, hydrogenerators and power transformers.

At the international level the problems of measurement methods for equipment noise has, so far, taken the form of general standards for the measurement of machine noise and specific (international, national or corporative) standards for rotating electric machines, internal combustion engines, pneumatic equipment, fans. The problem of limiting values has been tackled especially for rotating electric machines by the CEI and the CAER recommendations, corporative norms (VDE 0530-11-72) and national standards (for example, GOST 16372-70 and STAS 8278-68).

Conclusions

1. The equipment that runs in industrial plants constitutes the principal cause of the noxious noise level in work spaces.

2. At the present time, normalization of equipment noise takes the form especially of international, national and corporative norms concerning the methodology of measuring equipment noise (general norms and specific norms).

3. The norms concerning the admissible limiting levels for equipment that have been worked out up to the present time are aimed only at ensuring safe operating conditions (correct execution), for which reason they can become the basis of legislation for the manufacture of equipment that generates a noxious noise level. /337

4. Concern with equipment noise must be broadened and deepened by initiating studies aimed at reducing it to values that are not noxious for man.

5. The manufacture of silent equipment will be generalized on the basis of the results of soundproofing studies (point 4) by working out standards of admissible limiting noise levels for equipment.

6. The operating conditions attributed to equipment in norms of measurement and limiting levels of equipment noise must correspond to concrete operating conditions, and the measurement methods

and units employed in measuring noise must be compatible with those employed for the admissible limiting noise levels of the sanitary norms.

Summary

The present study shows the importance of equipment noise normalization in the complex of measures aimed at reducing noise in work spaces, as well as the necessity of correlating these norms with the criteria that establish the noxious values of noise for man. /338

ACHIEVEMENTS IN NOISE AND VIBRATION CONTROL OF SHIPS BUILT IN RUMANIA

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Summary

Shipboard noise and vibrations due to the running of internal /339 combustion engines and other assemblies and machines in a small space, have a negative effect on acoustic comfort as well as the running of steering and control mechanisms.

The present study shows our concern with the reduction of noise and vibrations onboard ships built in Rumania, with two practical examples.

Thus, the present work describes the methodology of measuring and interpreting vibration amplitudes onboard ships during trial runs and offers solutions for noise reduction onboard 10,000/12,500 tdy prototype ships; the study also shows the efficiency of using inertial and elastic connecting points between lining and metal body of the control room in the machine compartment onboard the 300-passenger riverboat built by the Oltinita Shipyard.

NOISE ABATEMENT PROBLEMS IN THE POLISH IRON INDUSTRY

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Summary

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Intense work has been under way in the field of noise abatement for the last few years in the Polish iron industry. It includes protection for both present and future plants and objectives. In general, the work includes the following:

-- soundproof encapsulation of noise sources (machinery, installations, pipelines, fittings).

-- mufflers for expanding gases and vapors.

-- the use of sonically insulating cranes*.

-- noise abatement in rolling mill facilities.

-- noise abatement in rolling mill facilities [sic].

-- noise abatement in installations for the preparation and transport of raw material.

-- sound-absorbing ceilings for work areas.

-- soundproof compartments and control rooms.

*[Translator's note: Last word uncertain; typographical error in original.]

METHOD FOR PRACTICAL DETERMINATION OF SUBWAY NOISE

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Summary

On the basis of dimensional considerations concerning the train and noise measurement in a free field according to normalized methods, it is possible with the aid of a graph to determine the train's noise level at a point of space free of any obstacle. /343